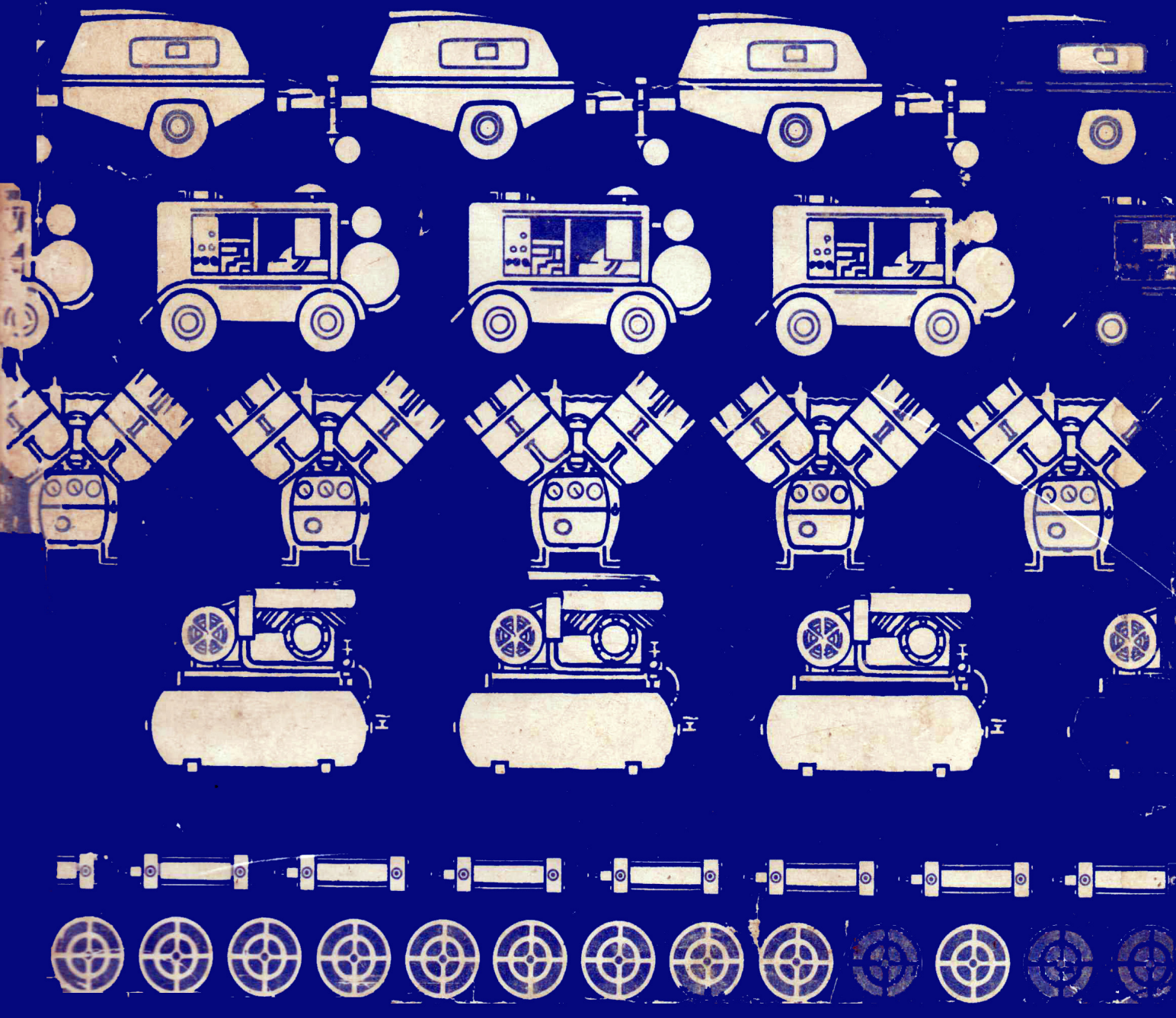
Pneumatic Handbook

6th edition



2
25
37

SUNNY EN[illegible]PRISES (PAK)
411 FA[illegible] [illegible]ENT[illegible]E BLOG-2,
RAJA GHAZ[illegible] FAR A.I ROAD,
P. O. BOX 15527, KARACHI-4.

PNEUMATIC HANDBOOK

6TH EDITION

by R.H.Warring

TRADE & TECHNICAL PRESS LTD.,
MORDEN, SURREY SM4 5EW,
ENGLAND.

ISBN 85461 090 1

©TRADE & TECHNICAL PRESS LIMITED. 1982

All rights reserved. No part of this publication may be reproduced, stored in a retrieval system, or transmitted in any form or by any means, electronic, mechanical, photocopying, recording or otherwise, without the prior written permission of the publishers, Trade & Technical Press Limited, Crown House, Morden, Surrey.

Printed in Great Britain by TTP (Printers) Ltd., East Molesey, Surrey

PREFACE

IN THIS era of energy conservation, stimulated by diminishing natural resources and the escalating costs of discovering, retrieving and processing natural and synthetic fuels for industry, compressed air is the most cost-effective, safest and simplest means of producing and transmitting power for industry. There is no consumption of vital minerals, no hazards to life or limb, no high capital investment, no waste disposal problems, no transportation costs, no political or environmental constraints. There is a limitless and ubiquitous supply of basic material – air. In these respects it is a unique source of power, quickly produced, easily controlled, infinitely variable, highly adaptable and low in cost.

The PNEUMATIC HANDBOOK proves conclusively the very wide and varied attributes of compressed air and pneumatic engineering, dealing fully with the subject from principles, through compression, selection, comparison, installation, conservation, actuation, control, techniques, equipment, application, and known national and international standards and data. There is no more comprehensive and practical tome of knowledge to be found in one volume at this time. Already it has been translated and published in French, Italian, Spanish and Portuguese. It could be termed the 'bible' of Pneumatics.

The editorial content of this 5th Edition has been compiled and edited over a period of twelve months from the latest information available or submitted by the various experts, authorities and organizations, to whom due acknowledgment is given, with particular appreciation to the British Compressed Air Society, CETOP, PNEUROP and its Members. The very latest information and data on trends and traits in the future of the Pneumatic Industry of Europe is to be found in the monthly technical journal "Power", produced by the Publishers of this Handbook.

The Publishers

schrader bellows

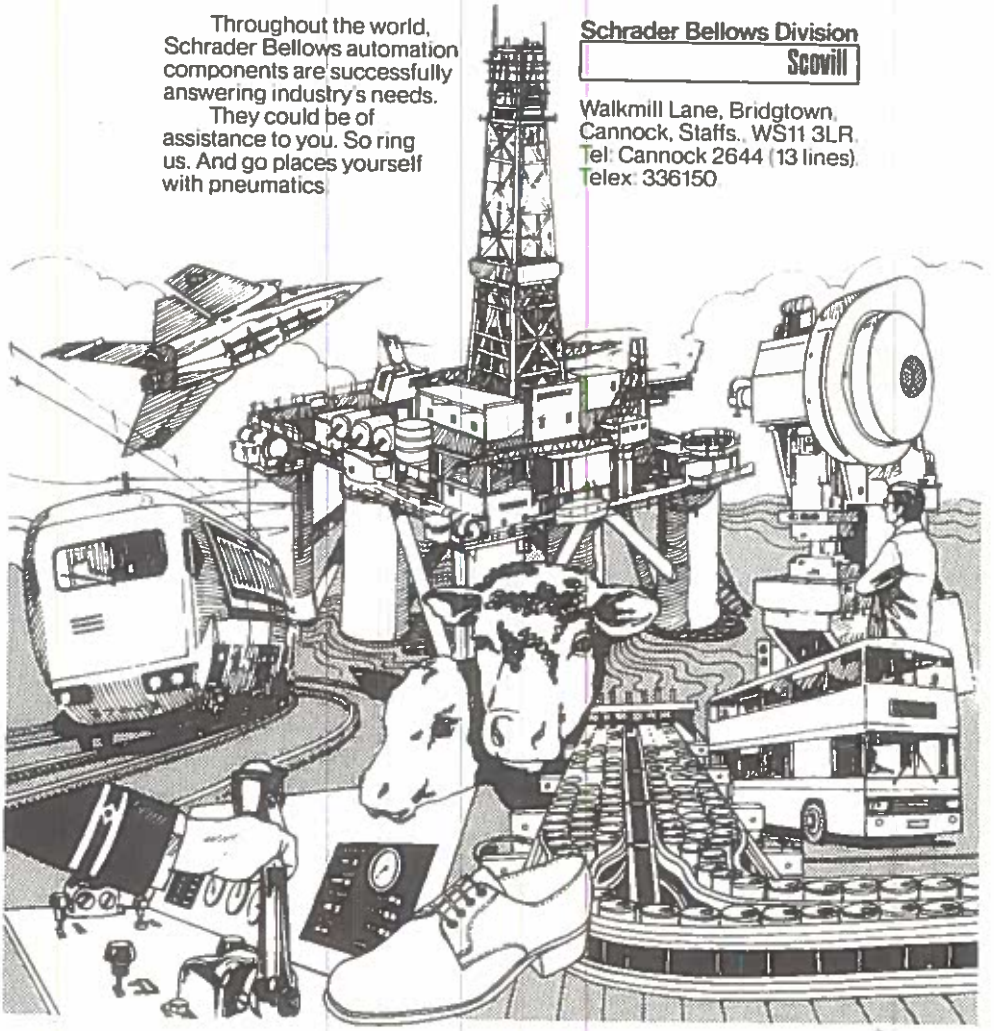
GOING PLACES WITH PNEUMATICS

Throughout the world, Schrader Bellows automation components are successfully answering industry's needs.

They could be of assistance to you. So ring us. And go places yourself with pneumatics.

Schrader Bellows Division
Scovill

Walkmill Lane, Bridgtown,
Cannock, Staffs., WS11 3LR.
Tel: Cannock 2644 (13 lines).
Telex: 336150



CONTENTS

SECTION 1 — Principles

Terms and Definitions 1
Properties of Air and Gases 5
Gas Laws and Thermodynamics 15
Compressible Gas Flow 17
Pneumatics and Mechanization 28
Pneumatics and Automation 33
Safety 43
Cost Effectiveness 50

SECTION 2A — Compressors

Compressors 59
Compressor Performance 73
Compressor Noise Suppression 82
Compressor Lubrication 85
Compressor Controls 89
Compressor Installation 97
Air Receivers 102
Heat Exchangers and Aftercoolers 113
Compressor Acceptance Tests 119
Compressor Selection 124

SECTION 2B — Air Lines

Compressed Air Mains 135
Branch Lines 143
Pressure Drop Calculations 146
Pipe Materials 153
Compressed Air Treatment 158
Air Line Fittings 174

SECTION 2C — Valves

Valves 181
Valves Actuation 192

SECTION 2D — Actuators

Air Cylinders 197
Air-Hydraulic Cylinders 217
Actuators. 221
Air Motors 224

SECTION 2E — Ancillaries

Seals 237
Pressure Gauges. 244
Air Flow Measurement 250

SECTION 3

Circuits 259
Air-Hydraulic Circuits 268
Logic Circuits 271
System Maintenance 278

SECTION 4A — Tools

Pneumatic Tools 289
Contractors' Tools. 297
Workshop Tools 306
Foundry Equipment 317
Spray Techniques 320

SECTION 4B — Devices and Machines

Lift, Hoists and Winches 325
Air Starters 328
Air Springs. 331
Pneumatic Timers 337
Pneumatic Sensors. 340
Compressed-Air Aspirators 343
Turboexpanders 345
Air Blowers 347
Airmovers 354
Vacuum Pumps. 356

SECTION 5 — Techniques

Noise Control 367
Low Temperature Pneumatics 373
High Pressure Pneumatics 376
Vacuum Techniques. 384

WE DIDN'T GET WHERE WE ARE TODAY JUST BY MAKING SUPERLATIVE PNEUMATICS PRODUCTS. IT'S THE SUPERLATIVE SERVICE, TOO.

The Sperry Vickers pneumatics products range is available through our own branches and an excellent distribution network throughout Britain.

It is based on ISO and CETOP standards and includes a comprehensive choice of advanced spool, poppet and slide valves and 'MIDAS' power cylinders.

Equally available is the advice and expertise of Sales Engineers to help you with problem solving; circuit design; cost-cutting and trouble-shooting. Also our new facility for supplying custom built control panels.

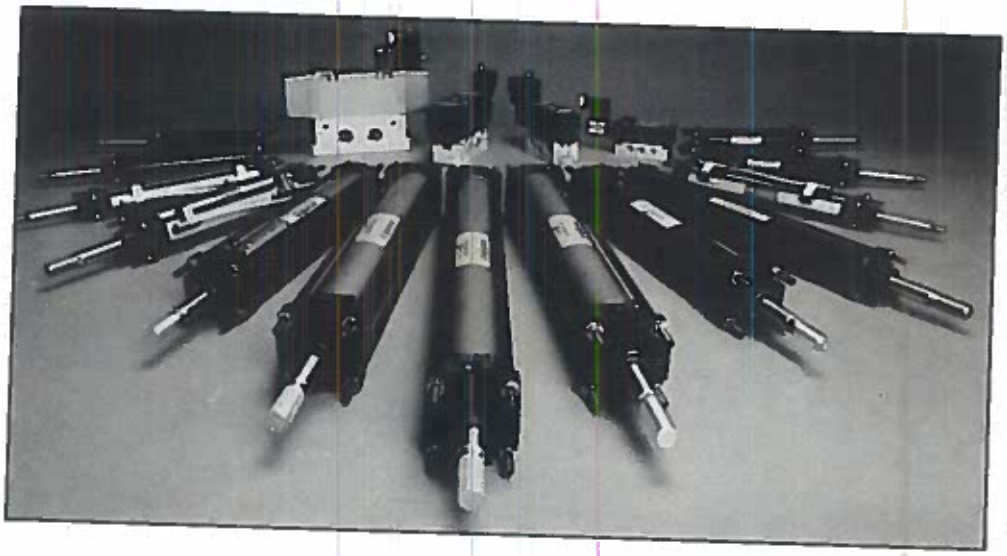
And to familiarise you with our products and their application potential there is detailed literature – catalogues, data sheets and instruction manuals.

Sperry Vickers products and service facilities reflect the unremitting quest for excellence pursued by this great company with its international back-up and development resources.

Thinking of tomorrow puts us where we are today – No. 1, in automation and pneumatics.

SPERRY VICKERS

Sperry Vickers, Automation & Pneumatics, Halesfield 6, Telford, Shropshire. Tel: 0952 586000 Tlx. 35286



With international standardisation becoming more important, Joucomatic now has over 200 stocking sales points in the World, among a comprehensive range of pneumatic control equipment are products complying with both ISO and CETOP standards including:-

- 540/541 pneumatic and solenoid spool valves.
- Subbases ISO 5599 sizes 1 – 2 – 4.
- The range is completed by a mini spool valve dia. 4 which also has the same technical advantages as the ISO range.
- Use on lubricated air – unlubricated air or dry air.
- Special non-stiction design.
- Tie rod cylinders dia. 32 to 200 mm TRINORM.
- To CNOMO, ISO and CETOP standards – cylinders for lubricated or unlubricated air can be equipped with magnetic detection.

A range of round cylinders dia. 12 to 25 mm, single acting and double acting to ISO/CETOP standards.

- A complete range of accessories, solenoid valves, poppet valves, size range M5 to 1½" BSP.
- JOUCOMATIC – an international distribution network servicing the whole World (literature available on request).

JOUVENEL ET CORDIER
B.P. 312,
92506 Rueil Malmaison Cedex,
France.
Tel: 010 331 749 0360
Telex: Jouco 203661

JOUCOMATIC CONTROLS LTD,
Air Automation House
Navigation Street
Wolverhampton WV1 3RF
Tel: Wolverhampton (0902) 59924 (4 lines)
Telex: 337372 JGLAA

GO- WITH P&V
GREEN TOOLS
Now it's easy to pick the best in Contractors and Industrial tools. They're GREEN
GREEN = P & V = QUALITY
P & V, with years of experience, are painstaking in choice of steel. Expert in forging and heat treating. Meticulous in finishing and inspecting.
Result – quality. An overworked word, given its true meaning by P & V.
Here's another P & V first – new skin wrapped "Handipacks" for the range of smaller tools.
Go for quality. Go for P & V. Go for GREEN.
At your P & V distributor now!
P&V
Padley & Venables Ltd.
Cavlywhite Lane, Dronfield
Sheffield S18 6XY, England
Tel: Dronfield (0246) 413301
INTL + 44 246 413301. Telex: 54254
Write or phone for our latest product literature and details of authorised distributors.
COOPER INDUSTRIES

We pioneered the pneumatic push-in fitting system, developed the largest range and ensure it's ex stock. No imitators can match us.

Introduced to the market in 1970, the LF 3000 range continues to be expanded to include innovative devices of benefit to our customers.

Just look at the range and that's not all of them. Before you buy a system that cannot expand with you, better find out why the major pneumatics companies insist on fitting the LF 3000. For full details, write to

Legris Limited, Ashville Trading Estate, The Runnings, Cheltenham, Glos. GL51 9PT or telephone (0242) 43103.

Sizes – Metric 4 to 14 mm OD tube (inch units available for companies that have not yet converted to metric).
Threads – BSP taper or parallel. Also M5.
Pressure – Maximum working 18 BAR.

legris
LF 3000

Pneumatic & hydraulic fittings and ball valves

GRINDING...SANDING...CHIPPING...TAPPING...DRILLING...

SCREWDRIVING...NUTRUNNING...FASTENING...POLISHING...

LIFTING...PUSHING...PUMPING...DISPENSING...SPRAYING...

PAINTING...WASHING...GLUEING...RUSTPROOFING...

CONTROLLING...LUBRICATING...REGULATING...FILTERING...

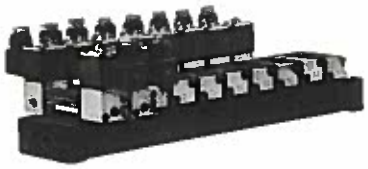
ARO AIR PRODUCTS

THE TOOLS FOR THE JOB

ARO

THE ARO CORPORATION (U.K.) LTD.
EAST MOONS MOAT INDUSTRIAL ESTATE
ALDERS DRIVE, REDDITCH, WORCS. B98 0RH
TEL : 25722. TELEX 338169

Control Problems Solved!



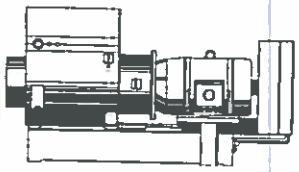
Leading manufacturers of electro-pneumatic control equipment - including explosion protected solenoid valves

RGS

Further details and full range Catalogue from:—
R.G.S. Electro-Pneumatics Limited
Church Street, Accrington, Lancashire
Telephone: Accrington 35015. Telex: 63425
'Everything under Control'

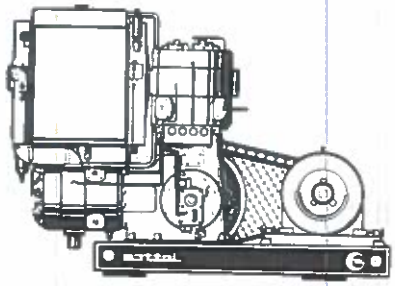
compressed air

aria compressa



Motor Driven Rotary Compressors

elettrocompressori



«L» Range Reciprocating Square Compressors

compressori a squadra

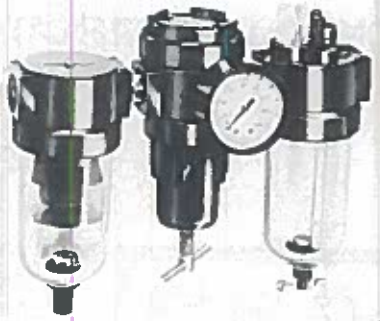
Diesel Driven Rotary Compressors

motocompressori

ING. ENEA MATTEI SpA
Strada Padana Superiore, 307
20090 VIMODRONE (Milano)
Tel. (02) 2500441 (5 linee ric. aut.)
Telegrammi: MATTEICOM
Telex: 321193 COMAVI I

mattei

You've seen many unique features, now Parker combine them all.



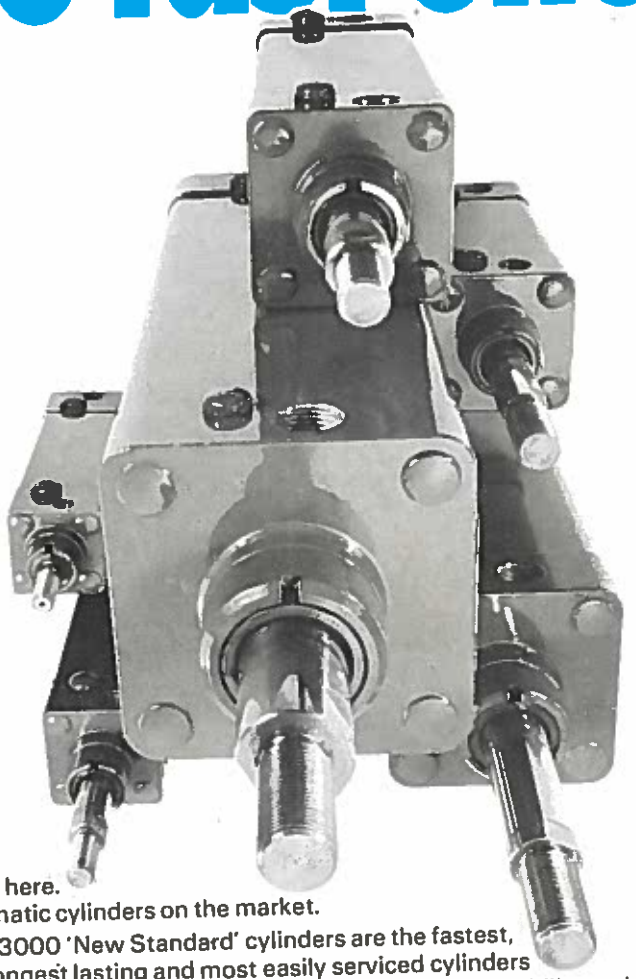
Parker air preparation equipment:–

Robust
Reliable
Competitive

Parker
Fluidpower

Parker Hannifin
P.O.Box 170, Parker House,
6, Greycaine Road,
Watford, WD2 4QA, England.
Tel: Watford (0923) 44377
Telex: 23765

The fast ones.



Now! They're here.
The fastest pneumatic cylinders on the market.

The new M/13000 'New Standard' cylinders are the fastest, best cushioned, longest lasting and most easily serviced cylinders available. They offer the benefits of faster work rates, greater reliability and less downtime. At prices competitive with ordinary cylinders this all adds up to profit.

Designed entirely to BS.4862:1973 and C.E.T.O.P. RP43P, they are available with a full range of mountings and stocked stroke lengths.

For further details contact us today.

Tomorrow's standard today.

Martonair

Martonair Ltd., 128 St. Margarets Road, Twickenham TW1 1RJ. Telephone: 01-892 4411. Telex 935562

AIR LINE SERVICE UNITS

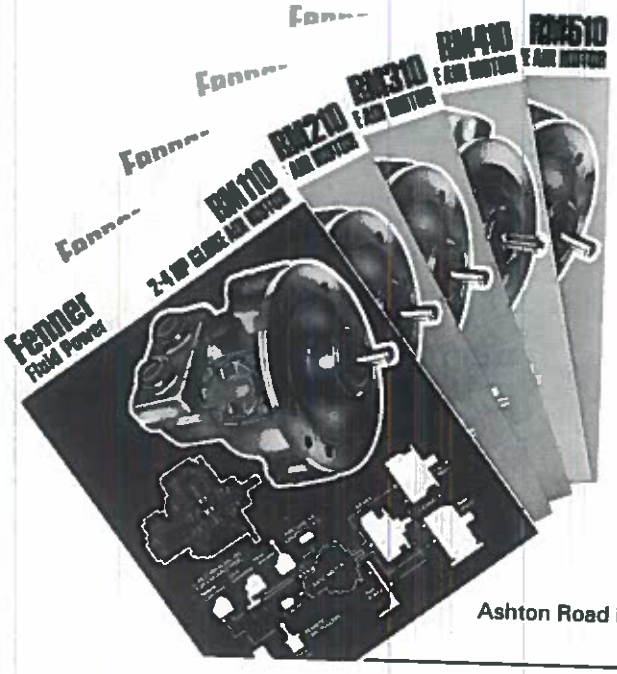
A RANGE OF AIR LINE FILTERS/ REGULATORS/LUBRICATORS, AND INTEGRAL FILTER/REGULATOR

Sizes — 1/8" to 2.1/2" B.S.P.

High Efficiency and performance characteristics, flow rates up to 1000 cfm and pressures of 200 p.s.i. ensure that our range is imminently suitable for OEM or general plant installations.

Filters and Lubricators fitted with polycarbonate bowls, as standard. Metal bowl guards or metal bowls available, if required.

WATTS FLUID POWER LIMITED
London Road, Thrupp, Stroud,
Gloucestershire, England.



Air Mail

Globe air motors deliver — fast. Drop us a line and we'll mail you the very latest information on the first class range.

Fenner
Fluid Power

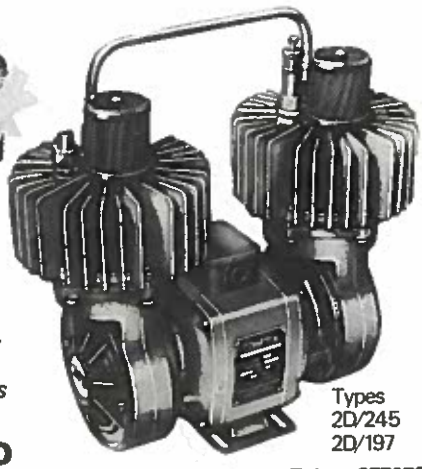
Division of J.H. Fenner & Co. Ltd.
Ashton Road Harold Hill Romford RM3 8UA Essex
Telephone (04023) 43851

COMPTON COMPRESSORS

COMPRESSION, VACUUM AND CIRCULATION APPLICATIONS

Type D/351 VM

Type D/351 VM MX-2



Types 2D/245 2D/197

A wide range of units is available all incorporating the unique Compton diaphragm principle, completely sealing the compressor unit and ensuring absolutely oil-free operation.

Send for our illustrated brochure which gives full details of the Compton range or consult us with your particular requirements.

DAWSON, McDONALD & DAWSON LTD

Compton Works, Ashbourne, Derby DE6 1DB. Tel: Ashbourne (0335) 43184 Telex: 377072

TTP PRINTERS LTD

MAGAZINES

HOUSE JOURNALS

DESIGN ORIGINATION

BLACK & COLOUR PRINTING

COMPETITIVE PRICES

PROMPT SERVICE

ESTIMATES WITHOUT OBLIGATION

For further details contact:—

13/15 CREEK ROAD
EAST MOLESEY
SURREY
KT8 9BE
ENGLAND
TELEPHONE: 01 941 2100

MIDLAND PNEUMATIC

The Control Experts

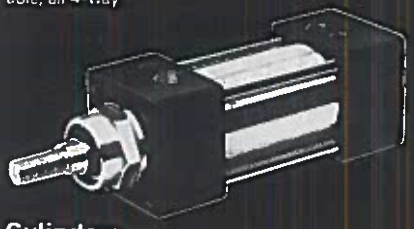
Your pneumatic control problem won't worry us. We've probably solved something similar before and with the most up to date equipment available. Our range of valves, cylinders and air line fittings is extensive.

Compact control valves

A new range of slide valves incorporating the patented 'Fluoslide' feature giving low frictional resistance. ⅛" and ¼" B.S.P. Fully metricated. Pilot air or solenoid operation with manual over-ride.

Hand control valves

Semi rotary in ¼" and ⅜" B.S.P. sizes. Brass body with 'Fluoslide' P.T.F.E. lapped slide. Various types available, all 4-way.



Cylinders

Metric and Imperial bore cylinders available. Brass or hard anodised aluminium barrels. Stainless steel piston rods. Heavy duty end covers.

Midland Pneumatic Ltd
Wednesfield, Wolverhampton,
WV11 3DZ, England
Telephone 0902 732501/2

Compared to our couplings,

Drallim ¼ OD Tee Coupling — Actual Size

others can be a huge expense.

Competitor's ¼ OD Tee Coupling — Actual Size

Our tube couplings are exceptionally compact — among the smallest fittings on the market — and they feature the unique Drallim captive seal.

Made of neoprene rubber, the captive seal prevents tube movement or vibration and ensures a leaktight connection — even under the most severe mechanical conditions.

We've designed these couplings for applications at pressures up to 10 Bar, with either copper or plastic tubing. They simplify tube assembly, save space, and connection can be re-made with ease any number of times.

What's more, as the seal is held captive within the nut there's no olive to get lost or damaged.

All of which adds up to some very useful economies over conventional larger fittings.

Our captive seal couplings are available in stainless steel, brass or plastic — in metric and imperial sizes. Send for full information about these and the many other couplings in our range.

DRALLIM

DRALLIM TUBE COUPLINGS LTD
Brett Drive, Bexhill-on-Sea, East Sussex TN40 2JR
Tel (0424) 216611. Telex: 95285.

Air Films 390
Air Bubble Techniques 392
Pneumatic Conveyors 394
Compressed Air for Breathing 398

SECTION 6 — Data

Standards and Publications 403
Graphical Symbols for Pneumatic Systems and Components 408

SECTION 7

Survey of Compressors 437

SECTION 8

Buyers' Guide 459
Editorial Index i
Advertisers' Index xiv

SECTION 1

Principles

Terms and Definitions

THE FOLLOWING terms and definitions apply in the pneumatics industry, arranged in alphabetical order for convenience of reference (but with closely related subjects grouped so that their inter-relationship is clearly established).

Absolute pressure	see *Pressure.*
Absolute temperature	see *Temperature.*
Adiabatic process	compression of a gas under conditions of no heat exchange with the surroundings.
Aftercooling	removal of heat from the air after compression is completed.
Booster	a machine or device for taking in air (or gas) and delivering it at a higher pressure.
Capacity	(compressor)actual volume rate of flow compressed and delivered at the discharge point at stated inlet conditions.
Clearance volume	the volume remaining in a compression space in a cylinder at the end of a compression stroke.
Compressibility factor (Z)	a factor giving the deviation of the behaviour of a real gas compared with an ideal gas.
Compression ratio	ratio of final pressure to original pressure.
Condensate	liquid formed by condensation of water vapour in air due to a fall in temperature.
Displacement	actual swept volume, or volume displaced by a compressing device per stroke or per stage.
Displacement compressor	compressor working on the principle of compressing air by displacement of a member (*eg* a piston in a cylinder).
Dynamic compressor	rotary machine converting kinetic energy into pressure energy.

Filter	a device for removing contaminants from a fluid.
Free air	air at atmospheric condition at the inlet point of a compressor.
Ideal compression	conditions appertaining to the isentropic compression of an ideal gas.
Ideal gas	a gas which conforms exactly to the theoretical gas laws.
Intercooling	the cooling of compressed air (or gas) between compression stages.
Isentropic efficiency	ratio of real gas isentropic power consumption to shaft inlet power.
Isentropic power consumption	theoretical power required to compress a gas under constant entropy.
Isothermal power consumption	theoretical power required to compress a gas at constant temperature.
Packaged compressor	a complete, powered compressor unit ready to operate.
Polytropic process	compression or expansion of a gas where the relationship $P.V^n$ = a constant applies, the exponent n having various values from 1 to 1.43.
Pressure — absolute	pressure measured from absolute zero.
atmosphere	absolute pressure of the atmosphere.
dynamic	velocity pressure or total pressure minus static pressure.
gauge	pressure measured above atmospheric pressure.
static	pressure measured under static conditions for the air or gas (*ie* not influenced by velocity).
total	pressure measured at a stagnation point when a moving air-stream is brought to rest. With stationary air or gas, equals the static pressure.
vacuum	difference between atmospheric pressure and a (lower) value of absolute pressure.
Pressure ratio	ratio between absolute discharge pressure and absolute inlet pressure.
Pressure regulator	a valve or similar device for reducing line pressure to a lower (constant) level.
Pulsation damper	a chamber designed to eliminate pulsations present in air flow.
Relative clearance volume	ratio of clearance volume to volume swept by a compressing element.
Relative humidity	ratio of water vapour actually present in air to the saturated water vapour content, at a given temperature and pressure.

Separator	a device for removing liquids (*eg* water and oil) from compressed air.
Specific energy requirement	input energy requirement per unit of compressed air produced; or shaft input power per unit of compressor capacity.
Specific power consumption	specific energy requirement.
Stage pressure ratio	compression ratio for any particular stage in a multi-stage compressor.
Surge limit	the flow limit below which operation of a dynamic compressor becomes unstable.
Temperature — absolute	temperature above absolute zero in Kelvin.
ambient	temperature of the environment.
discharge	total temperature at the standard discharge point of a compressor.
inlet	total temperature at the standard inlet point of the compressor.
total	temperature at the stagnation point of an airstream.
Volumetric efficiency	ratio of capacity to displacement of a compressor or vacuum pump.

NOTES ON UNITS

The following units are now generally accepted as standard in the compressed air industry in Europe.

Pressure — *bar* (1 bar = 10^5 N/m^2 = 10^5 Pa)

Linear — *metre* (m) for dimensions of 1 metre or greater.
— millimetre (m) for dimensions less than 1000 mm.

Volume — *metre*^{3} (m^3) is the standard unit for volumes

or

Capacity — above 1000 litres
— *litre* (l) is the standard unit for volumes below 1000 litres.
Compressor and air tool manufacturers prefer litres.
Pneumatic equipment manufacturers prefer dm^3.
Both have the same numerical value (1 litre = 1 dm^3).
The smaller volume unit cm^3 is used in fluidics.

Velocity — metres/second (m/s)

Rotational Speed — rev/min (rpm) continues to be widely used, although rev/sec (r/s) is the more correct unit.

Mass — kilogram (kg), or tonne (t).

Lifting capacity — tonne (t).

Work, Energy
or Heat — joule (J).

Power — watt (W) or kilowatt kW.

Note: brake horsepower (bhp) is now regarded as obsolete and if referred to is replaced by *brake power*.

Torque — Newton.metre (N.m).

Specific power consumption — kilowatt-seconds/m^3 (kW.s/m^3)
or J/dm^3.

Stress — Newtons/m^2 (N/m^2), or N/mm^2.

Note: the Pascal (Pa) is also used by the high vacuum industry (1 N/mm^2 = 1 MN/m^2 = 1 MPa).

Flow — litres/second (l/s).

Note: for a rough approximation l/s divided by 2 = ft^3/min;
for a close approximation 1 l/s = 0.47 ft^3/min, and 1 ft^3/min = 2.1 l/s.

Mass flow — kilograms/second (kg/s).

Moisture content of air or gases — kg/m^3, or g/m^3.

Properties of Air and Gases

CLEAN, DRY air is a mechanical mixture of approximately 78% by volume nitrogen and 21% oxygen, the remaining 1% being made up with minor quantities of some fourteen other gases. The *composition* of air remains substantially the same between sea level and an altitude of about 20 kilometres, but its *density* decreases with increasing altitude, and varies with pressure and temperature. At sea level, with a pressure of 1 bar and a temperature of 15°C, the density of air is 1.208 kg/m^3 (0.0765 lb/ft^3). Thus 1 kg of air has a volume of 0.818 m^3; or 1 lb of air a volume of 13.07 ft^3.

At a specific temperature and pressure, the number of molecules in a unit volume is constant for any gas or mixture of gases. For a temperature of 0°C and a pressure of one atmosphere, this number is 2.705×10^{19} molecules per cubic centimetre, or 7.84×10^{23} molecules per cubic foot.

At standard temperature and pressure, the mean velocity of gas molecules is of the order of 500 metre/second, with a mean free path between intermolecular collision, for air, of the order of 3×10^{-6} millimetres. The rate of collision under such conditions is responsible for the pressure exerted by air (or any gas) on a surface immersed in it, or on walls containing the gas. Specifically, therefore, the pressure of any gas depends on its mass density, the number of molecules present and their mean velocity.

The effect of a change in temperature is to modify the value of the mean velocity. Hence, in the absence of any other change, the resultant pressure will vary with the temperature. Similarly, any change in volume will effectively modify the mass present and again affect the pressure. Thus pressure, temperature and volume are interrelated. This relationship can be expressed in the following form for a perfect gas (which assumes that the molecules are perfectly elastic, are negligible in size compared with the length of their mean free path, and exert no force on each other);

$$PV = RT$$

where

P = pressure
V = volume
T = absolute temperature
R = gas constant

In engineering units

$$PV = RT \times 10^{-5}$$

where

P is in bar

V is the specific volume of gas in m^3/kg

R is the gas constant in J/(kg.K) = 287.1 J/(kg.K) for air

Atmospheric Air

Atmospheric air normally contains water vapour, and the total pressure of the air is the sum of the partial pressures of the dry air and water vapour. The air is saturated when the partial pressure of the water vapour is equal to the saturation pressure of the water vapour at that temperature. The saturation pressure is dependent only on the temperature (see also Table I and Fig 1).

TABLE I — VAPOUR PRESSURE OF MOIST AIR

(Pressure in millibars)

Temp. °C	Relative Humidity in Percent									
	10	20	30	40	50	60	70	80	90	100
−10	0	1	1	1	1	2	2	2	2	3
−5	0	1	1	2	2	2	3	3	4	4
0	1	1	2	2	3	4	4	5	5	6
5	1	2	2	3	4	5	6	7	7	8
10	1	2	4	5	6	7	9	10	11	12
15	2	3	5	7	9	10	12	14	15	17
20	2	5	7	9	12	14	16	19	21	23
25	3	6	10	13	16	19	22	25	29	32
30	4	8	13	17	21	25	30	34	38	42
35	6	11	17	22	28	34	39	45	51	56
40	7	15	22	30	37	44	52	59	66	74
45	10	19	29	38	48	57	67	77	86	96
50	12	25	37	49	62	74	86	99	111	123

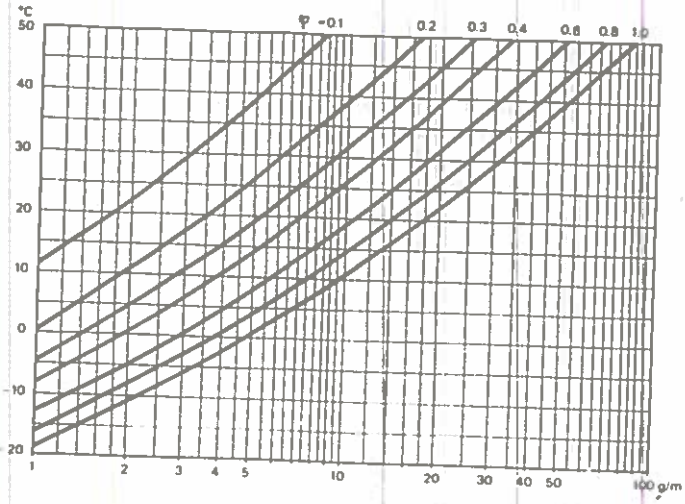


Fig 1 Water content of air at different relative humidities

When air is cooled at constant pressure, the *dew point* is reached, when the partial pressure is equal to the saturation pressure. Any further cooling will then result in water separating by condensation (see also Table II).

Water Content of Moist Air

The water vapour content of moist air can be determined from the following equation derived from the gas laws

$$m_w = \frac{Q \cdot P_s \cdot V_t}{R_w \cdot T} \times 10^5$$

where

m_w = mass of water vapour in kg
Q = relative humidity (expressed as a decimal fraction)
P_s = saturation pressure of water vapour in bar at given temperature (see Table II).
V_t = total volume of air/water vapour in m^3
R_w = gas constant for water vapour
= 461.3 J/(kg.K)
T = absolute temperature of air/water vapour mixture in K (= degrees C + 273.2)

The gas constant of the air/water vapour mixture can also be determined as:

$$R_m = \frac{(287.1\ m_w) + (461.3\ m_w)}{(m_a + m_w)}$$

when

m_a = mass of dry air in kg

From this the density of the air/water vapour mixture follows as

$$\text{density } (\rho) = 10^5 \times \frac{P_s}{R_m \cdot T} \text{ kg/m}^3$$

Compressibility

For many engineering purposes the assumption that real gases conform to the perfect gas equation is satisfactory. The basic equation may, however, readily be extended to express the deviation of a real gas from a perfect gas when subject to compression, by introducing a compressibility factor (Z) when

$$PV = ZRT$$

TABLE II – SATURATION PRESSURE p_s AND DENSITY, ρ_w, OF WATER-VAPOUR AT SATURATION
VALUES BELOW 0°C REFER TO SATURATION ABOVE ICE

t °C	p_s mbar	ρ_w g/m³
−40	0.128	0.119
−38	0.161	0.148
−36	0.200	0.183
−34	0.249	0.225
−32	0.308	0.277
−30	0.380	0.339
−29	0.421	0.374
−28	0.467	0.413
−27	0.517	0.455
−26	0.572	0.502
−25	0.632	0.552
−24	0.699	0.608
−23	0.771	0.668
−22	0.850	0.734
−21	0.937	0.805
−20	1.03	0.884
−19	1.14	0.968
−18	1.25	1.06
−17	1.37	1.16
−16	1.51	1.27
−15	1.65	1.39
−14	1.81	1.52
−13	1.98	1.65
−12	2.17	1.80
−11	2.38	1.96
−10	2.60	2.14
− 9	2.84	2.33
− 8	3.10	2.53
− 7	3.38	2.75
− 6	3.69	2.99
− 5	4.02	3.25
− 4	4.37	3.52
− 3	4.76	3.82
− 2	5.17	4.14
− 1	5.62	4.48
0	6.11	4.85
1	6.57	5.19
2	7.06	5.56
3	7.58	5.95
4	8.13	6.36
5	8.72	6.80
6	9.35	7.26
7	10.01	7.75
8	10.72	8.27
9	11.47	8.82
10	12.27	9.40
11	13.12	10.01
12	14.02	10.66
13	14.97	11.35
14	15.98	12.07
15	17.04	12.83
16	18.17	13.63
17	19.37	14.48
18	20.63	15.37
19	21.96	16.31
20	23.37	17.30
21	24.86	18.34
22	26.43	19.43
23	28.09	20.58
24	29.83	21.78
25	31.67	23.05
26	33.61	24.38
27	35.65	25.78
28	37.80	27.24
29	40.06	28.78
30	42.43	30.38
31	44.93	32.07
32	47.55	33.83
33	50.31	35.68
34	53.20	37.61
35	56.24	39.63
36	59.42	41.75
37	62.76	43.96
38	66.26	46.26
39	69.93	48.67
40	73.78	51.19
41	77.80	53.82
42	82.02	56.56
43	86.42	59.41
44	91.03	62.39

Compressibility charts may be used to determine specific values of Z for any particular gas or gas mixture. Alternatively a generalized chart may be used, devised on the basis that the behaviour of all gases is similar when their reduced pressures (p_R) and reduced temperatures (t_R) are the same, *ie*

$$p_R = \frac{P}{P_c} \quad \text{and} \quad t_R = \frac{T}{T_c}$$

where

P_c = critical pressure and

T_c = critical temperature

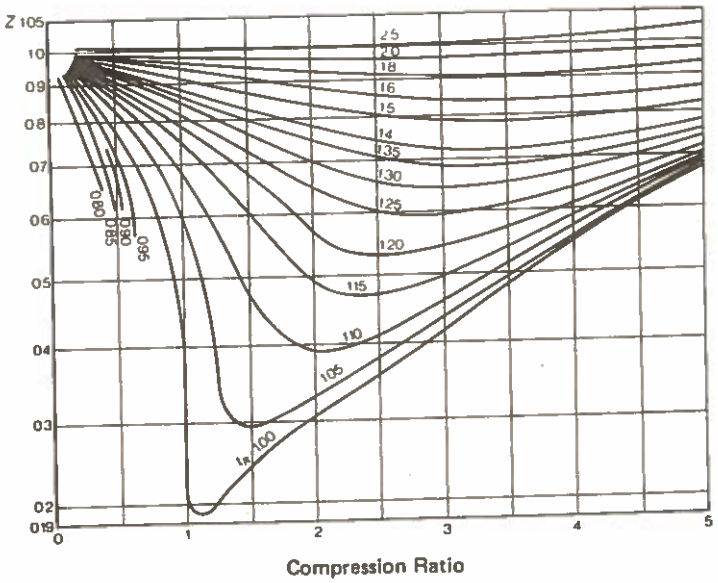


Fig 2 Generalized compressibility factor Z

Such a chart is shown in Fig 2. To use, values of the critical constants pR and tR are determined on a proportionate basis and summed, *eg*

$$p_R \text{ (gas 1)} = \frac{P}{P_C \text{ (gas 1)}} \times A\%$$

$$p_R \text{ (gas 2)} = \frac{P}{P_C \text{ (gas 2)}} \times B\%$$

and so on, where A and B are the percentage proportions expressed as a decimal fraction.

Then, finally, $p_R = p_R \text{ (gas 1)} + p_R \text{ (gas 2)} + \ldots\ldots$

The reduced temperature (t_R) of the mixture is determined in a similar way. The corresponding value of the compressibility factor can then be read off the chart. Table III gives critical pressure and critical temperature values for a wide range of gases.

Air Compression

The four possible ways of compressing air (or any gas) are shown in Fig 3. Of these, the first two involve either the removal or addition of heat and are not directly connected with pneumatic engineering which is primarily concerned with pressure/volume relationship and the general gas equation

$$P \cdot V = \text{a constant}$$

TABLE III – PHYSICAL DATA FOR SOME GASES

Gas	Formula	Molecular mass	Gas constant J/(kg·K)	P_{cr} bar (abs)	t_{cr} °C	Normal boiling point °C	c_p kJ/(kg·K)	c_p/c_v
acetylene	C_2H_2	26.04	319.4	62	36	−83	1.659	1.26
air	—	28.96	287.1	38	−141	−194	1.004	1.40
ammonia	NH_3	17.03	488.2	113	132	−33	2.093	1.30
argon	Ar	39.94	208.2	49	−122	−186	0.522	1.67
benzene	C_6H_6	78.11	106.5	49	289	80	1.728	1.08
butane, N-	C_4H_{10}	58.12	143.1	38	152	−1	1.635	1.10
butane, iso-	C_4H_{10}	58.12	143.1	36	135	−12	1.620	1.10
butylene	C_4H_8	56.11	148.3	39	144	−7	1.531	1.11
carbon dioxide	CO_2	44.01	188.9	74	31	−78	0.833	1.30
carbon disulfide	CS_2	76.13	109.2	75	278	46	0.657	1.20
carbon monoxide	CO	28.01	296.6	35	−140	−191	1.039	1.40
carbon tetrachloride	CCl_4	153.84	54.0	44	283	77	0.833	1.18
chlorine	Cl_2	70.91	117.3	77	144	−34	0.498	1.35
cyan	$(CN)_2$	52.04	159.8	61	128	−21		1.26
dichloromethane	CH_2Cl_2	86.95	95.6	103	216	41		1.18
ethane	C_2H_6	30.07	276.5	49	32	−89	1.714	1.19
ethyl chloride	C_2H_5Cl	64.51	128.9	53	185	−12		1.13
ethylene (ethene)	C_2H_4	28.05	296.4	51	9	−104	1.515	1.24
Freon 12	CCl_2F_2	120.92	68.8	39	112	−30	0.611	1.13
Freon 13	$CClF_3$	104.46	79.6	39	29	−82	0.644	1.15
Freon 14	CF_4	88.01	94.5	37	−46	−128	0.894	1.16
Freon 22	$CHClF_2$	86.47	96.1	49	96	−41	0.646	1.20
helium	He	4.00	2077.2	2	−268	−269	5.208	1.66
hexane, N-	C_6H_{14}	86.17	96.5	30	234	69	1.617	1.06
hydrogen	H_2	2.02	4124.5	13	−240	−253	14.270	1.41
hydrogen bromide	HBr	80.92	102.8	82	90	−88		1.42
hydrogen chloride	HCl	36.46	228.0	83	52	−85	0.792	1.41
hydrogen cyanide	HCN	27.03	307.6	49	184	−89		1.27
hydrogen sulphide	H_2S	34.08	244.0	90	101	−59	0.996	1.33
hydrogen jodide	HJ	127.93	65.0	–	151	−36		1.40
krypton	Kr	83.80	99.3	55	−64	−153	0.249	1.68
methane	CH_4	16.04	518.3	46	−83	−162	2.205	1.31
methyl ether	$(CH_3)_2O$	46.07	180.5	52	127	−23		1.11
methylene chloride	CH_2Cl_2	84.94	97.9	63	238	40	0.826	1.18
methyl chloride	CH_3Cl	50.49	164.7	67	143	−24	0.826	1.25
neon	Ne	20.18	412.0	27	−229	−246	1.032	1.66
nitrogen	N_2	28.02	296.8	34	−147	−196	1.038	1.40
nitric oxide	NO	30.01	277.1	65	−94	−152	0.990	1.39
nitrous oxide	N_2O	44.02	188.9	73	37	−89	0.874	1.30
oxygen	O_2	32.00	259.8	51	−118	−183	0.915	1.40
ozone	O_3	48.00	173.2	69	−5	−112		1.29
pentane, N-	C_5H_{12}	72.14	115.3	34	197	36	1.624	1.06
pentane, iso-	C_5H_{12}	72.14	115.3	33	187	28	1.601	1.06
propane	C_3H_8	44.09	188.6	43	97	−42	1.620	1.13
propylene (propene)	C_3H_6	42.08	197.6	44	92	−48	1.521	1.15
sulphur dioxide	SO_2	64.07	129.8	79	158	−10	0.607	1.25
water (superheated)	H_2O	18.02	461.4	221	374	100	1.860	*)
xenon	Xe	131.30	63.3	58	17	−108		1.66

*) for wet steam: $c_p/c_v \approx 1.035 + 0.1 \cdot X$

where X is the steam content ($X > 0.4$)

for superheated steam (up to 10 bar and 800°C):

$c_p/c_v \approx 1.332 + 0.112 \cdot 10^{-3} \cdot t$

where t is the temperature in °C

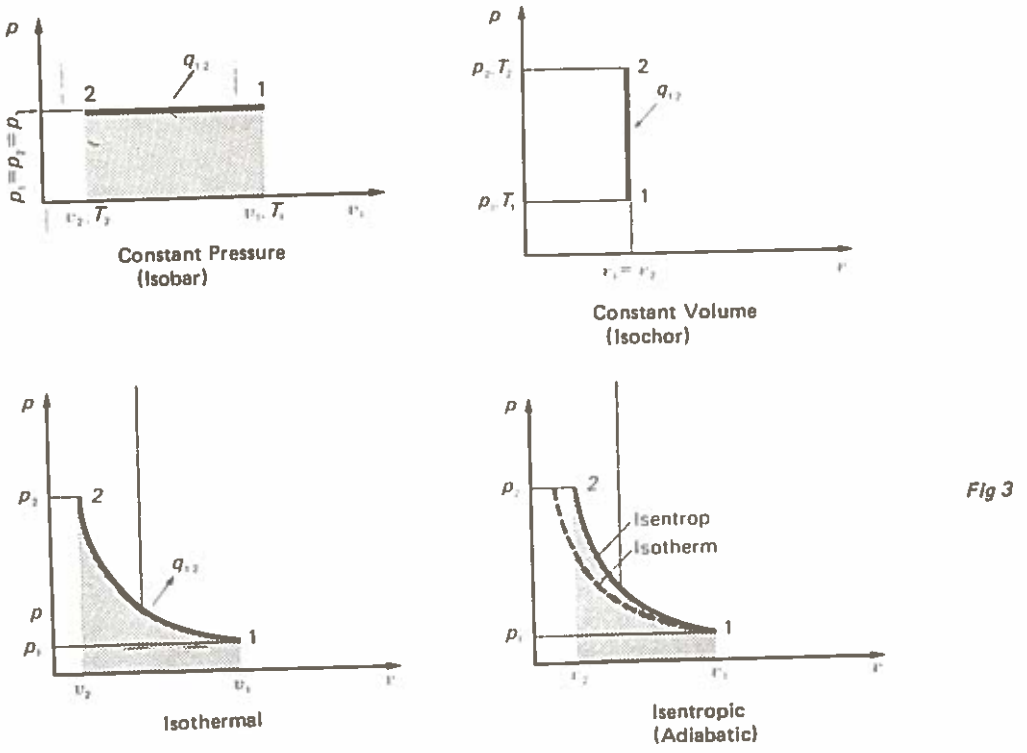


Fig 3

This, in fact, is consistent with *isothermal* compression where the process takes place at constant temperature, when pressure ratios and volume ratios have a direct reciprocal relationship

$$P_2/P_1 = V_1/V_2$$

To achieve this, cooling must be present to take away an amount of heat equal to the work of compression, *ie*

heat to be lost = $m \cdot R \cdot T_1 \log_e (P_2/P_1)$ joules

or = $m \cdot P_1 V_1\log_e (V_1/V_2) \times 10^5$ joules

where pressures P_1 and P_2 are absolute pressures in bars

m = mass of air (or gas) in kg

V_1 and V_2 are specific volumes in m^3/kg

T_1 is the initial (and constant) temperature in deg K.

With *adiabatic* compression (also called *isentropic* compression) there is no heat exchange with the surroundings, *ie* no cooling present, and so the rise in pressure and reduction in volume are

accompanied by a rise in temperature of the air (or gas). The pressure/volume relationship is

$$P_2/P_1 = (V_1/V_2)^k$$

or $$P \cdot V^k = \text{constant}$$

where k is the ratio of the specific heats

The change in temperature from T_1 to T_2 is given by

$$P_2/P_1 = (T_2/T_1)^{\frac{k}{k-1}}$$

or the temperature of the compressed air (or gas) T_2 is given directly by

$$T_2 = T_1 (P_2/P_1)^{\frac{k-1}{k}}$$

In practice neither isothermal or isentropic compression is likely to apply, but the process will be somewhere between the two or *polytropic.* The general equation then becomes

$$PV^n = \text{constant}$$

where the value of the exponential n lies between 1 and 1.405 for air

and compressed air temperature $T_2 = T_1 (P_2/P_1)^{\frac{n-1}{n}}$

Compression Efficiency

Compression efficiency can be defined as the ratio of the theoretical work required to compress a given amount of air to a specific volume, to the actual amount of work done by the compressor.

Since the units of pressure are F/l^2, and those of volume are l^3, work done can be expressed by the product of pressure and volume (units F x l).

The mean effective pressure (mep) with isothermal compression is then

$$\text{mep} = P_1 \log_e \frac{P_2}{P_1}$$

$$= P_1 \log_e r$$

where r is the compression ratio

The isothermal indicated horsepower or *Isothermal Factor* is given by

$$K \cdot P_1 \log_e r = K \cdot P_1 \log_e \frac{P_2}{P_1}$$

K being a constant factor, depending on the units employed.

More generally (or in metric units), the work for isothermal compression is given as

$$\text{Work (joules)} = 10^5 \, P_1 \cdot V_1 \log_e \frac{P_2}{P_1}$$

where
P is the absolute pressure in bar
V is the volume in m^3

For isothermal compression from atmospheric pressure

$$\begin{aligned}\text{Work (joules)} &= 10^5 \, V_1 \log_e P_2 \\ &= 230.3 \, V_1 \log_e P_2 \\ &\quad 2.303 \, V_1 \log_{10} P_2\end{aligned}$$

The power for isothermal compression follows by replacing volume flow in m^3 by m^3/sec, viz

$$\text{power (kW)} = 230.3 \, V_F \log_{10} / P_2$$

where
V_F = volume flow m^3/sec
P_2 = final pressure in bar

Power for *adiabatic compression* can be determined in a similar manner only in this case the mean effective pressure is

$$\text{mep} = \frac{k}{k-1} \times P_1 \left(r^{\left(\frac{k-1}{k}\right)} - 1 \right)$$

$$= 3.463 P_1 \, (r^{0.29} - 1)$$

Thus power required for adiabatic compression is given by:

$$\text{power (kW)} = 2.303 \, V_1 \log_{10} \left(\left(\frac{P_1}{P_2} \right)^{0.29} - 1 \right)$$

Equivalent Free Air Volume

The relationship between free air volume and compressed air volume follows directly from the compression ratio, viz:-

$$\frac{\text{free air volume}}{\text{compressed air volume}} = \text{compression ratio}$$

or

$$\frac{V_{FA}}{V_C} = r$$

whence

$$V_{FA} = \frac{P_2 + P_1}{P_1} \times V_C$$

where

P_2 = final pressure, absolute

P_1 = atmospheric pressure

For P_2 in lb/in^2

$$V_{FA} = \frac{P_2 + 14.7}{14.7} \times V_C$$

See also chapters on *Compressor Performance* and *Compressible Air Flow*.

Gas Laws and Thermodynamics

THE GAS laws are based on the behaviour of perfect gases or mixtures of perfect gases. They are:-

Boyle's law – which states that the volume (V) of a gas, at constant temperature, varies inversely with pressure (P).

$$V_2/V_1 = P_1/P_2$$

or

$$P_1.V_1 = P_2.V_2 = \text{a constant}$$

Clarke's law – which states that the volume of a gas, at constant pressure, varies directly with absolute temperature (T).

$$V_1/V_2 = T_1/T_2$$

or

$$\frac{V_1}{T_1} = \frac{V_2}{T_2} = \text{a constant}$$

Amaton's law – which states that the pressure of a gas, at constant volume, varies directly with absolute temperature

$$P_1/P_2 = T_1/T_2$$

or

$$\frac{P_1}{T_1} = \frac{P_2}{T_2} = \text{a constant}$$

Dalton's law – which states that the total pressure of a mixture of gases is equal to the sum of the partial pressures of the gases present (partial pressure being the pressure each gas would exert if it alone occupied the same volume as the mixture).

$$P = \sum_{1}^{n} p_1$$

Avagado's law — which states that equal volumes of all gases under the same conditions of pressure and temperature contain the same number of molecules.

Amagat's law — which states that the volume of a mixture of gases is equal to the partial volumes which the constituent gases would occupy if each existed alone at the total pressure of the mixture.

Poisson's law — which states that for a process without any heat exchange with the surroundings the relationship between pressure and volume follows the mathematical relationship

$$P_1 . V_1^{\,k} = P_2 . V_2^{\,k}$$

where

k is the ratio of the specific heat of the gas at constant pressure to specific heat at constant volume

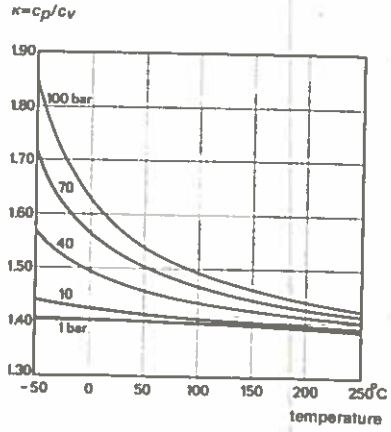


Fig 1 The isentropic exponent for air.

The value of the exponential k varies with temperature and pressure (see Fig 1) but for pressures of less than 1 bar (and all temperatures)

k = 1.40 for air
= 1.66 for mono-atomic gases
= 1.40 for two-atomic gases
= 1.30 for three-atomic gases

General Gas law — resulting from a combination of Boyle's law and Clarke's law.

$$P.V/T = \text{a constant}$$

Laws of Thermodynamics

(i) For any system of constant mass the amount of work done on or by the system is equal to the amount of energy transferred to or from the system.

(ii) Energy exists at various temperature levels but is available for use only if it can pass from a higher to a lower temperature level.

Compressible Gas Flow

AIR, STEAM and gases are all *compressible fluids* and the d'Arcy equation used for determining pressure and head losses with liquid flow is no longer applicable because the density of gases and vapours changes considerably with changes of pressure. For simplified general engineering calculations not requiring great accuracy, liquid (d'Arcy) flow formulas may be used if the pressure drop involved is less than 10% of the inlet pressure. Use of such formulas is also sometimes extended to pressure drops up to 40% of the inlet pressure, provided in this case the specific volume is taken as the average of the upstream and downstream conditions.

The real flow of a pressurized gas through pipes differs appreciably in a number of important characteristics from the flow of liquids in pipes. Pressure, for example, drops at an increasing rate along the pipe, rather than with a constant pressure gradient. At the same time velocity tends to increase up to a maximum defined by $V = \sqrt{kgRT}$ for air, but subject to a limiting or maximum length, which must correspond to the end of the pipe. At this point the pressure gradient is infinite, *ie* the pipe is effectively closed. In this equation k is the ratio of specific heats at constant pressure to constant volume, R the individual gas constant, and T the absolute temperature. An alternative formula is $M = \sqrt[1]{k}$ (= 0.845 for air), where M is the Mach number.

The general equation may be written in the same form as the d'Arcy equation for fluid flow, but with the addition of an extra term representing the pressure drop required to increase the flow momentum:

$$\frac{\Delta P}{L} = \frac{f}{D} \cdot \frac{\rho V^2}{2} + \beta \rho \bar{Q} \frac{dV}{dL}$$

where ΔP is the pressure drop
L is the pipe length
f is a constant friction factor (but dependent on surface roughness)
D is the pipe diameter
ρ is the gas density
$\bar{Q}$ is the specific volume of gas
dV/dL is the velocity gradient
β is a factor of the order of unity and normally taken as 1.0 (*ie* can be eliminated from the equation).

Actual flow conditions may range from adiabatic to isothermal. Adiabatic conditions are only likely to apply in short, well insulated pipes where no appreciable heat is transferred to or from the pipe. Isothermal flow or flow at constant temperature is commonly assumed as more consistent with normal practice, especially for long pipes. In fact, most practical pipelines will generate polytropic flow conditions, virtually impossible to analyze. The assumption of isothermal flow is thus a practical compromise.

Isothermal Flow

With isothermal flow, a formula developed from basic principles is

$$\frac{\Delta P}{L} = \frac{\frac{f}{D} \cdot \frac{\rho V^2}{2}}{kM^2 - 1}$$

The friction factor is dependent on the Reynolds number of flow and pipe roughness. It can be assumed independent of Mach number.

The Reynolds number will be constant for isothermal flow, but may vary with adiabatic or isentropic flow (and certainly with diabatic flow), in which case a mean value can be assumed.

The Reynolds number is a dimensionless quantity, given arithmetically by

$$R_e = \frac{\rho VD}{\mu}$$

$$= \frac{VD}{\nu}$$

where

ρ is the mass density
μ = viscosity
ν = kinematic viscosity

} in units consistent with velocity (V) and pipe diameter (D)

For air at standard temperature $\nu = 1.45 \times 10^{-5}$ in^2/sec

The friction factor (f) is common for all fluids (*ie* gases and liquids) and is normally determined from empirical charts. For laminar flow, the friction factor is dependent only on Reynolds number and is numerically equal to $64/R_e$, laminar flow being defined by the Reynolds number not exceeding 2000. For turbulent flow and smooth bore pipes, the Reynolds number can be calculated from the following empirical formula –

$$f = \frac{0.3164}{R_e^{0.25}}$$

As the Mach number (M) approaches 0, the denominator in this equation comes closer and closer to unity, reducing the equation to the same as that for liquid flow. There is thus some justification for using the d'Arcy formula for compressible flow calculations (as mentioned initially) as M→0 (*ie* consistent with short lengths of pipes with resulting low pressure drops). This also implies that at

very low Mach numbers, compressible flow can be treated as incompressible. Pressure gradients ($\Delta P/L$) will, in fact, be within 5% of incompressible flow values at Mach numbers up to about 0.18 for air.

A complete working formula for isothermal flow is:

$$V_m^2 = \left[\frac{144gA^2}{\bar{Q}\left(\frac{fL}{D} + 2\log\frac{P_1}{P_2}\right)}\right]\left[\frac{P_1^2 - P_2^2}{P_1}\right]$$

where

A	=	cross sectional area of pipe, ft^2
g	=	acceleration of gravity = 32.2 ft/sec^2
L	=	length of pipe, feet
f	=	friction factor
D	=	pipe bore, feet
P_1	=	absolute pressure (entry)
P_2	=	absolute pressure (exit)
V_m	=	mean velocity of flow, ft/sec
$\bar{Q}$	=	specific volume of gas, ft^3/lb

Limiting Values

Maximum possible velocity (V^*) in a pipe is source velocity (M=1), given directly by

$$V^* = 11.32\sqrt{kg\,P\bar{Q}}$$

where

V^* is in ft/sec

k is the ratio of specific heats of the gas

P is the absolute pressure in bar

$\bar{Q}$ is the specific volume of gas, m^3/kg

Maximum pressure (P^*) can be determined on the basis of continuity, viz:

$$V_1 P_1 = VP = V^* P^*$$

Hence

$$\frac{P^*}{P_1} = \frac{V_1}{V^*} = \frac{M_1}{M^*} = M_1\sqrt{k}$$

where the suffix 1 refers to initial conditions

Hence (and since the velocity of sound is constant under isothermal conditions)

$$P^* = P_1 . M_1 . k$$

Limiting Length (L^*) can be determined from the general equation

$$\frac{fL^*}{D} = \frac{1}{kM_1^2} - 1 - \log_e \frac{1}{kM_1^2}$$

or

$$L^* = \frac{D}{f}\left(\frac{1}{kM_1^2} - 1 - \log_e \frac{1}{kM_1^2}\right)$$

The implication of this is that velocity of gas flow can go on increasing in a pipe up to a maximum Mach number of $1/\sqrt{k}$. This increase ceases at a limiting length of pipe (L^*) which must be at the end of the pipe. If the actual length of pipe is greater than L^*, initial conditions will have to be adjusted to reduce M_1 so that the actual pipe length is less than, or equal to, L^*.

At the same time the limiting pressure ratio (P^*/P_1) and limiting length (L^*) are dependent only on initial velocity (Mach number) and the k value of the gas.

Pressures and lengths between two sections of a pipeline can thus be expressed as follows, knowing the Mach numbers at each section:

$$\frac{P_2}{P_1} = \frac{M_1}{M_2}$$

$$\frac{fL}{D} = \left(\frac{fL^*}{D}\right) M_1 - \left(\frac{fL^*}{D}\right) M_2$$

Adiabatic Flow

Similar treatment applies, although the corresponding formulas are more complicated and may require working as a series of approximations in order to reach real values in particular cases. In general, the pressure, temperature and velocity will always be slightly less than those for isothermal flow, but the limiting length will be similar. The differences are usually small enough to be negligible, except at higher Mach numbers, and thus, for simplification of calculations, isothermal formulas can be used for subsonic adiabatic flow.

Limiting Velocity:

This is given directly by:

$$\frac{V^*}{V_1} = \frac{1}{M_1}\sqrt{\frac{2\left(1 + \frac{k-1}{2} M_1^2\right)}{k+1}}$$

Limiting Temperature:

$$\frac{T^*}{T_1} = \frac{2\left(1 + \frac{k-1}{2} M_1^2\right)}{k+1}$$

Limiting Length:

$$L^* = \frac{D}{F}\left(\frac{1-M_1^2}{kM_1^2}\right) + \frac{k+1}{2k}\log_e\left(\frac{(k+1)M_1^2}{2\left(1+\frac{k-1}{2}M_1^2\right)}\right)$$

Limiting Pressure:

$$\frac{P^*}{P_1} = M_1\left(\frac{2\left(1+\frac{k-1}{2}M_1^2\right)}{k+1}\right)$$

Stagnation State

Flow is possible between two extremes. At one extreme, velocity is zero and temperature is a maximum, since all the kinetic energy is converted to enthalpy. The speed of sound is also a maximum (stagnation point or stagnation state). At the other extreme, the velocity is a maximum and the temperature falls to absolute zero, all the enthalpy being converted into kinetic energy. The speed of sound is then zero (zero temperature state). Between these extremes the practical flow may be subsonic, transonic, or supersonic — see Fig 1 — although the zero temperature state can never be reached (*ie* is a hypothetical condition).

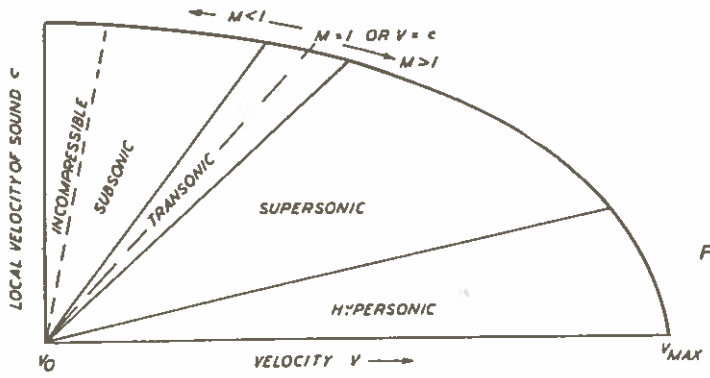


Fig 1

At the stagnation state

$$\frac{O^2}{2} + h_o = \frac{V^2}{2} + h$$

or

$$h_o = \frac{V}{2} + h$$

From the general gas relationship it follows that the stagnation temperature (T_o) is given by

$$T_o = T + \frac{V^2}{2sp}$$

where sp = specific heat at constant pressure

Alternatively,

$$\frac{T_o}{T} = 1 + \frac{k-1}{2} \times M^2$$

where M = Mach number = $\frac{V}{c}$

The stagnation pressure can be derived as

$$\frac{P_o}{P} = \left(\frac{T_o}{T}\right)^{\frac{k}{k-1}}$$

$$= \left(1 + \frac{k-1}{2} M^2\right)^{\frac{k}{k-1}}$$

This may be expanded in the form

$$P_o = P + \frac{\rho V^2}{2} \left(1 + \frac{M^2}{4} + \frac{2-k}{24} M^4 + \ldots\ldots\ldots\ldots\right)$$

This can be compared with the equation for incompressible flow

$$P_o = P + \frac{\rho V^2}{2}$$

The difference between these two equations represents the effect of increased gas density due to compressibility, generally termed the *compressibility factor.* Values of the compressibility factor range from unity at very low Mach numbers (where there are no compressibility effects), up to 1.276 as the Mach number approaches 1 (velocity approaches the speed of sound in the gas). Besides increasing the dynamic pressure of compressible flow, compared with incompressible flow, the rising value of the compressibility factor can also affect the flow velocity through ducts with varying area. The relationship between area and velocity changes is, in fact, a function of the local Mach number, and can be rendered in the form

$$\frac{dA}{dV} = \frac{A}{V} \quad (M^2 - 1)$$

With subsonic flow, a decrease in area produces an increase in flow velocity and vice versa (similar to incompressible flow) ie, area and Mach number changes are opposite. The flow velocity may be sonic only at a constant section. With supersonic flow, a decrease in flow area produces a decrease in flow velocity, and vice versa, ie, area and Mach number changes are the same.

Flow from Stagnation Conditions

Gas compressed and stored in a reservoir is essentially under stagnation conditions, where velocity is zero and the pressure and temperature are known (or can be determined). Where the reservoir is used as a supply, the velocity, temperature and pressure at any other section of flow are determined basically from the following relationships:-

Velocity at any arbitrary section:

$$V = \sqrt{2sp\,T_o \left(1 - \frac{P}{P_o}\right)^{\frac{k-1}{k}}}$$

Alternatively, for adiabatic flow, the velocity at any section can be determined from the temperature at that section

$$V = \sqrt{2sp\,T_o\,(1 - \frac{T}{T_o})}$$

Flow Rate

Flow rate can be determined as the mass flow, *ie*, mass flow = $VA\rho$ or directly as the product of V and A in numerically consistent units

$$\text{Dimensions are } \frac{L}{T} \times L^2 = \text{flow rate} = \frac{L^3}{T}$$

Pressure at any arbitrary section:

$$P = \frac{P_o}{\left(1 + \frac{k-1}{2} M^2\right)^{\frac{k}{k-1}}}$$

Temperature at any section:

$$T = \frac{T_o}{1 + \frac{k-1}{2} \cdot M^2}$$

At any (constant) section where the flow is sonic the flow conditions are described as critical, yielding a critical temperature (T^*) and a critical pressure (P^*), where

$$\frac{T^*}{T_o} = \frac{2}{k+1} \qquad \text{(Adiabatic or isentropic flow)}$$

$$\frac{P^*}{P_o} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$$ (Isentropic flow only)

Note : For air, where k = 1.4, the value of critical pressure is $\left(\frac{2}{2.4}\right)^{\frac{1.4}{0.4}} = 0.528$

That is, the critical pressure is 52.8% of P_o. Similarly the critical temperature can be calculated as 83.3% of T_o.

Critical Area:

The relationship between the critical area (A^*) or throat area where $\mu = 1$ and the area of any other section (A) is given by

$$\frac{A}{A^*} = \frac{1}{M} \frac{\left(1 + \frac{k-1}{2} \cdot M_2\right)^{\frac{k+1}{2(k-1)}}}{\frac{k+1}{2}}$$

$$= \frac{1}{M} \left(\frac{1 + 0.2\,M^2}{1.2}\right)^3 \quad \text{for air}$$

Nozzle Flow

Flow at the throat of a nozzle, supplied by a reservoir or similar source under stagnation conditions, will be sonic if the critical pressure is greater than the receiver pressure — see Fig 2a. This means that the flow will be critical. The flow velocity follows by relating to the critical temperature, where

$$\text{flow velocity} = 49\sqrt{T^*} \text{ ft/sec}$$

(The velocity of sound in air = $c = 49\sqrt{T}$ feet per second where T is the absolute temperature in degrees Rankine).

If the critical pressure is less than the receiver pressure, then the flow cannot be critical — Fig 2b. In this case the flow will be subsonic and the exit pressure will equal the receiver pressure. The temperature can be calculated from the general formula, or from

$$T_1 = T_o \left(\frac{P_1}{P_o}\right)^{\frac{k-1}{k}}$$

The velocity is likewise calculated from the general formula.

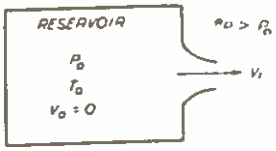


Fig 2a.

Fig 2b.

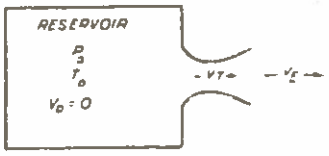


Fig 3

Similar analysis applies where the nozzle is of convergent-divergent form — Fig 3. In this case it is first necessary to establish whether the flow is critical or not (at the throat). The throat velocity can be determined accordingly; and from this the final exit velocity from the divergent section.

Flow through a nozzle can also be rendered directly in terms of flow rate and a discharge coefficient, this being a convention for engineering calculations. The complete nozzle formula is:-

$$\text{mass flow} = A \cdot E\,c \cdot \delta \cdot d^2 \sqrt{h}\ \sqrt{P_2/T}$$

where A is a constant depending on the units employed

E = coefficient for the velocity of approach

$$= \frac{1}{\sqrt{1 - m^2}}$$

$$\text{where } m = \frac{\text{cross sectional area of nozzle}}{\text{cross sectional area upstream}} = \frac{A_2}{A_1}$$

c = nozzle coefficient

δ = expansibility factor allowing for the change in air density which occurs during acceleration through the nozzle

$$= \frac{1 - 0.07h}{13.6\,P_2}$$ for values of m circa 0.16

(and h in inches wg and P_2 in inches of mercury)

d = diameter of nozzle

h = pressure drop across nozzle

P_2 = absolute pressure on downstream side of nozzle

T = absolute temperature on downstream side of nozzle

If T is in °R, P_2 in inches of mercury, h is in inches wg and d is in inches, a value of A = 0.1148 gives the mass flow in units of lb/sec.

For a specific nozzle profile, the formula can be simplified by the use of a nozzle constant appropriate to that particular geometry and nozzle size. Rendered as a solution for conventional flow rate (Q)

$$Q = K\,(T_1/P_1)\sqrt{h}\ \ \sqrt{P_2/T_2}$$

where

K = nozzle constant
T_1 = absolute temperature at specified inlet point
T_2 = absolute temperature at nozzle or specified point downstream
P_1 = absolute pressure at specified inlet point
P_2 = absolute pressure at nozzle or specified point downstream
h = pressure drop across nozzle

Simplified Orifice Formulas

An orifice is a simple form of nozzle, formed by a circular hole cut in a thin flat plate. Flow can again be determined with reference to an empirical discharge coefficient, or orifice coefficient. This will be much lower than for nozzles because of the less streamlined flow but, due to the simpler form of the nozzle, will be less subject to variation. Thus nozzle coefficients may vary between 0.90 (or less) and 0.995, depending on size and geometry, whereas an orifice coefficient can be expected to be of the order of 0.61, regardless of size, and differing only if the orifice has a well rounded, as opposed to a sharp, entry.

Very much simplified formulas can therefore be applied to assess the discharge of air through orifices, and the following are generally satisfactory for straightforward engineering calculations:-

(1) For upstream pressures above 14.7 $lb/in^2 g$

$Q\ (ft^3/min)$ for sharp-edged orifice $= \dfrac{218 \times A \times P_u}{\sqrt{460 + T}}$ (a)

or $= \dfrac{172 \times d^2 \times P_u}{\sqrt{460 + T}}$

$Q\ (ft^3/min)$ for rounded entrance orifice $\simeq \dfrac{417 \times A \times P_u}{\sqrt{460 + T}}$ (b)

or $\simeq \dfrac{263.5 \times d^2 \times P_u}{\sqrt{460 + T}}$

(2) For upstream pressures below 14.7 $lb/in^2 g$

$Q\ (ft^3/min)$ for sharp-edged orifice $= \dfrac{210 \times A \times P_u}{\sqrt{460 + T}}$ (c)

or $= \dfrac{166 \times d^2 \times P_u}{\sqrt{460 + T}}$

$Q\ (ft^3/min)$ for rounded entrance orifice $\triangleq \dfrac{324 \times A \times P_u}{\sqrt{460 + T}}$ (d)

or $\triangleq \dfrac{255 \times d^2 \times P_u}{\sqrt{460 + T}}$

where

A = orifice area, square inches
P_u = upstream pressure, $lb/in^2 g$
d = orifice diameter, inches
T = upstream air temperature °F

Note that for an upstream air temperature of 60°F these formulas further simplify to:

1 (a) $Q\ (ft^3/min) = 11.9\ A \cdot P_u$

or $= 9.4\ d^2\ P_u$

(b) $Q\ (ft^3/min) \triangleq 18.3\ A \cdot P_u$

or $\triangleq 14.4\ d^2\ P_u$

2 (c) $Q\ (ft^3/min) = 11.5\ A \cdot P_u$

or $= 9.05\ d^2\ P_u$

(d) $Q\ (ft^3/min) \triangleq 17.7\ A \cdot P_u$

or $\triangleq 13.92\ d^2\ P_u$

Pneumatics and Mechanization

MECHANIZATION IS broadly defined as the replacement of manual effort by mechanical power. In all degrees of mechanization, the operator remains an essential part of the system, although with changing demands on physical input as the degree of mechanization is increased. Table I illustrates the basic 'degrees' of mechanization, taking as an example a simple machining operation, such as drilling.

Pneumatics is an attractive medium for low cost mechanization particularly for sequential or repetitive operations. Many factories and plants already have a compressed air system installed, and manual machines can be mechanized with only moderate capital expenditure. Where the compressed air services are extended, however, additional compressors may have to be installed to meet the demand.

Additionally, compressed air is capable of providing both the power or energy requirements and the control system (although equally pneumatic control systems may be economic and can be advantageously applied to other forms of power). The main advantages of an all-pneumatic system are usually economy and simplicity, the latter reducing maintenance requirements to a low level. It can also have outstanding advantages in terms of safety.

Every application is different to some extent, although all may employ the same basic elements. Optimum design solutions must therefore be based on a detailed study of the economics and competitive performance of alternative systems.

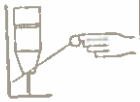
Power Units

The most widely used pneumatic component for power units is the *air cylinder.* The majority of air cylinders are 100 mm (4 in) diameter or smaller, giving a maximum thrust of the order of 450 kg (1000 lb) and a line pressure of 6 bar. Speed range usually allowed is from 0.5—2.5 m/sec (1.6—8 ft/sec), with speed control, if required, provided by throttling; more precise control can be obtained by connection with an oil-hydraulic cylinder.

Cylinders can be single- or double-acting; and are also capable of 'positioning' movements. As cylinders are basically linear actuators of limited power, it is essential to have methods of converting linear motion into semi-rotary motion, or of magnifying the force for actuating clamps, cutters, *etc.*

See chapters on *Air Cylinders* and *Air-Hydraulic Cylinders.*

TABLE I – BASIC DEGREES OF MECHANIZATION

Work Process		Work Effort and Work Rate	Operator's Responsibility	Comments
	HAND ONLY	Very limited	Effort, speed and accuracy	Also limited by hardness and thickness of material, etc. Accuracy depends on operator's skill rate
1.	HAND DRILL	Limited	Effort, speed and accuracy	Improved work capability rate and effort, also scope of operation. Accuracy still depends on operator's skill
2.	POWER DRILL	Work effort limited by acceptable weight of power tool	Part effort and accuracy, also work rate	Greatly improved work rate and scope of operation. Accuracy still depends on operator's skill
3. 	SIMPLE DRILL PRESS	Highest work effort needed. Work rate related to materials and subject	Mainly judgment, but also work rate	Similar to 2. with substantially less operator fatigue. Accuracy far less dependent on operator's skill
4.	SEMI-AUTOMATIC DRILL PRESS	As (3). Simultaneous multiple operation possible		Operator's role is largely supervisory

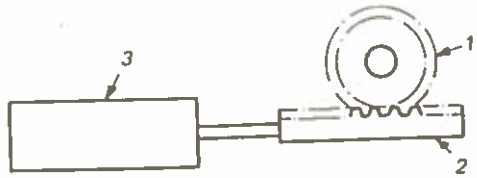


Fig 1 Rotating spindle is driven by a cylinder (3) through a limited number of revolutions by rack (2) and pinion(1).

Limited Rotary Movements

Where rotary outfits are required to work for not more than one or two revolutions, the use of an air cylinder is again a logical choice. A simple method of producing semi-rotary motion for one or two revolutions is by rack and pinion, as shown in Fig 1. The rack can be of the cut gear type, as shown, or formed by stretching roller chain along a flat support. Roller chain is also used in the method shown in Figs 2 and 3.

The method adopted will normally depend on the local conditions. An angular movement of 90° or less can also be obtained by a crank mechanism and this, by a ratchet and pawl, can be further extended to give an intermittently full rotary motion (Fig 4).

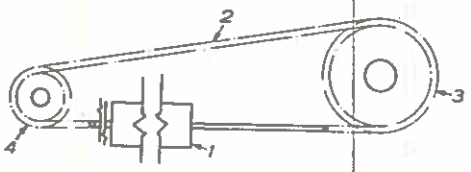


Fig 2 Rotary motion from a double rodded cylinder with chain drive. 1. Cylinder. 2. Chain 3. Pinion. 4. Idler pinion.

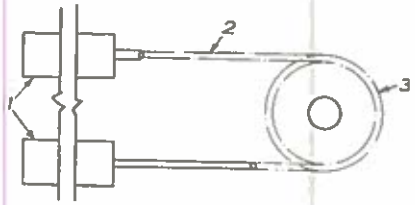


Fig 3 Rotary motion from two cylinders driving a chain pinion. 1. Single acting cylinder. 2. Chain. 3. Pinion.

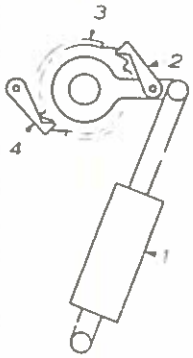


Fig 4 Ratchet drive giving intermittent uni-directional motion. 1. Cylinder. 2. Pawl. 3. Ratchet wheel. 4. Stop pawl.

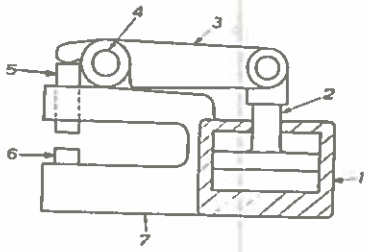


Fig 5 Small riveting machine with lever to multiply the force exerted by the cylinder. About 15 tons can be exerted over a short stroke. 1. Cylinder and frame. 2. Piston. 3. Lever. 4. Port. 5. Plunger. 6. Anvil. 7. Frame.

Force Magnification

This is best used when the final large force has only to be applied over a short distance. Then simple mechanical methods will give a limited degree of magnification. Fig 5 shows a simple squeeze riveting machine with a leverage of about 10:1. A toggle mechanism (Fig 6) has the advantage that a greater 'daylight' can be provided. The force exerted by a toggle mechanism at the end of the stroke is considerably influenced by the stiffness of the framework and links.

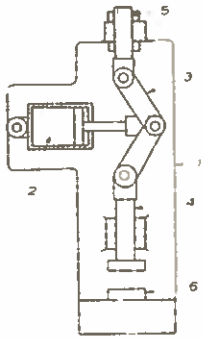


Fig 6 Toggle press. The force exerted by the plunger increases considerably as the toggle levers straighten out. By arranging the cylinder to 'pull' slightly more thrust is available for unlocking the toggle when commencing the upstroke. 1. Frame. 2. Cylinder. 3. Toggle. 4. Plunger. 5. Adjusting screw. 6. Anvil.

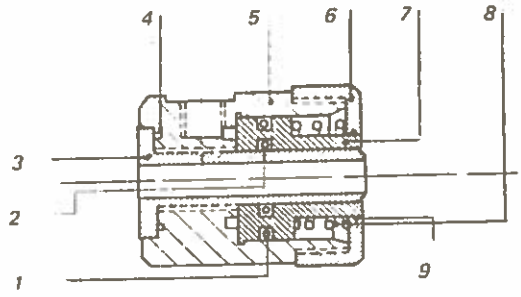


Fig 7 PowRlock clamping head for mounting on bolt or stud. 1. & 2. O-rings. 3. Spigot. 4. Seal. 5. Body. 6. Cap. 7. Piston. 8. Spring. 9. Wiper.

An air-hydraulic intensifier system has the advantage that line air pressure can be boosted to 175.7 bar (2500 lb/in^2) or so, thus allowing the use of very small actuators whilst retaining the advantages of pneumatic operation. Such a system can employ single shot air hydraulic intensifiers of various types, which are connected directly to one or more of the clamping heads. A typical head is shown in Fig 7. It is intended for fitting over a bolt or stud and provides an instant clamping action, so making spanners or screwed handles unnecessary. Up to 24 clamping heads can be operated simultaneously from one booster.

Linear Movements

The normal method is to arrange the cylinder so that it is co-axial with the part being moved, but this may not always be convenient and the use of levers and other devices may be justified. In this connection it may be pointed out that for a given length of stroke, a cylinder with piston extended will measure twice the stroke length plus 101–203 mm (4–8 in) to allow for seals, housings, *etc.* By using a shorter, larger diameter cylinder the stiffness is also greatly increased.

A simple lever (Fig 8) makes it possible to triple the cylinder stroke, but the length of movement depends on the space available for the lever. An alternative, using a bell crank, enables the

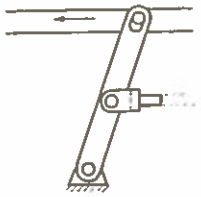


Fig 8 Simple lever to increase stroke.

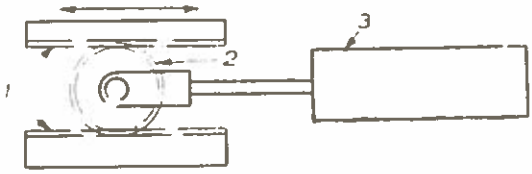


Fig 9 Doubling stroke length with stationary and moving racks. 1. Rack. 2. Pinion. 3. Cylinder.

cylinder to be placed at any angle. The traditional method of doubling the stroke using two racks (Fig 9) can also be useful. Racks of any length can be made by stretching roller chain along suitable supports.

To minimize the shock involved in accelerating and decelerating heavy slides, a method which involves a slight degree of magnification with approximate simple harmonic motion of the mass is shown in Fig 10. As the cylinder is dead centre at the ends of the stroke, two subsidiary cylinders are provided to give an initial impetus and if they are provided with suitable valving they can also help the final deceleration.

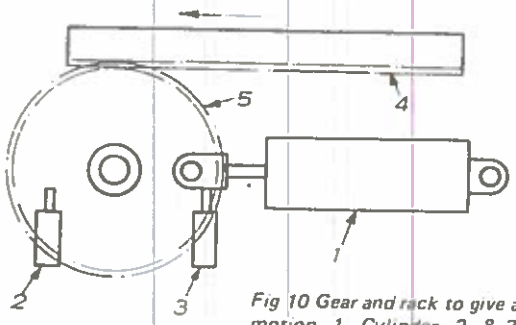


Fig 10 Gear and rack to give approximate harmonic motion. 1. Cylinder. 2. & 3. Subsidiary cylinder. 4. Rack. 5. Pinion.

Rotary Movements

Where continuous rotary movements are required at low to moderate power levels, the *air motor* becomes the logical choice and it is being used to an increasing extent in modern industry. *Vane-type* motors, characterized by a smooth power output are generally available in sizes up to about 1.5 horsepower. *Piston-type* motors are most widely used in the power range 1.5 to 15 horsepower. They work with lower rotary speeds than vane motors (usually under 5000 rev/min). Piston-type motors also have better starting and low speed characteristics than vane-type motors.

The usual working pressure for air motors is usually 6–7 bar (80–100 lb/in^2). See also chapter on *Air Motors*.

Industrial Robots

Widespread use is made of pneumatics for the mechanization of industrial robots, taking second place after hydraulics. In terms of percentage of robots using the different types of power systems, in current use hydraulics count for about 50%, pneumatics 30% and electrics 20%.

Power systems for industrial robots are simply advanced forms of mechanization rather than automation. The fact that industrial robots may be programmed, and thus work without attention from an operator, is reliant on a computer-type brain or microprocessor control unit taking the place of the human operator's brain. The power system of the industrial robot remains, essentially, a mechanical system, which may however incorporate varying degrees of sophistication *via* closed loop servo systems with feedback, and sensors. In this sense low technology robots can be described as coming under the heading of mechanization, and high technology industrial robots under automation.

See also chapter on Pneumatics and Automation.

Pneumatics and Automation

THE POINT at which 'mechanization' becomes 'automation' is where the machine operator's role is reduced to that of a supervisory nature, or even eliminated entirely. In other words, the essential feature of command is transferred to a timed operation of the system or semi- or fully-automatic control utilizing a suitable detector generating feedback signals.

Table I illustrates the basic degrees of automation ranging from a semi-automatic (basically *mechanized*) system through to fully programmed automation. True (full) automation starts when the operator is relieved of all responsibility.

Controls

The simplest control method is a manually operated control valve, which, even in an automated system, is normally retained for manual override, or as an emergency control. The manual control valve remains the usual solution when the control required is too simple to justify other control methods, although valves of this type, triggered by machine movements, may readily be adapted for automated working.

Time Control

A time control usually takes the form of a camshaft running at a steady speed which actuates small valves or electric switches at predetermined intervals. Normally it is used where the governing factor is the time taken by the process rather than that taken for the pneumatic cylinder to complete its travel. Sometimes an interlock with a limit detecting device may be necessary.

Sequential Control

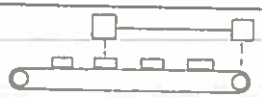
This is the normal method when two or more phases occur in an operation or series of operations. The end of each phase triggers off the next. Because of the difficulty of fixing the exact time taken for each phase, this method is usually preferred.

Sensing Devices

The choice of sensing device is bound up with the type of control system used, but the correct choice can, nevertheless, be very important and may, indeed, influence the control system.

Where great precision is required for a point of reversal, this can sometimes be done by using an air jet, as in pneumatic gauging.

TABLE I – BASIC DEGREES OF AUTOMATION

Category	Method		Basic System	Operator's Responsibility
Semi-automatic		Mechanically timed	Mechanical	Start/Stop, inspection and supervision
Semi-automation		Sequential control		Start/Stop, inspection and supervision
Automation		Sensing with feedback	Electro-mechanical	Start/Stop. Action on alarm signal
Full Automation		Programmed control	Programmed control by punched tape or computer	None required
Fully Programmed Automation		Fully programmed control	Computer controlled and capable of self-analysis, correction, etc.	None required

Electrical Control

Practically everything that can be done with electrical controls can be done pneumatically. However electrical controls may be preferred because of their compatibility with signals coming from electrical sources (*eg* timers); because wiring is neater than piping; or because of the convenience of microswitches to initiate control signals.

Using microswitches, proximity switches *etc*, with appropriate ancillary devices, in conjunction with double solenoid valves, very simple circuits can often be obtained. More complex control circuitry can be designed on intuitive lines, or by logic. The more complex the system, *eg* the greater the number of sequences required, the less competitive pneumatic control systems become compared with electronics, particularly with the introduction of low cost microprocessors and logic control units (LCU's).

Control System Design

Control system design must start with an analysis of the movements which must be performed. This analysis is usually done by means of a distance-time diagram. It is then necessary to allocate power components (*eg* cylinders) to achieve these movements and select suitable control components (*eg* valves). The major problem is then to plot the circuit, connecting the components to achieve correct working. This circuit plotting demands considerable skill and experience on the part of the designer.

Shift Register Modules

Shift register modules can provide a simplified approach to circuit design for pneumatic sequence controls. Basically each module is an in-line memory element (Fig 1). Modules are then interconnected in such a way that a switching step is performed only when its turn comes up. The number of individual movement steps indicate how many shift register modules are needed. The specific number of shift register modules are then lined up side by side to produce the required sequence or shift register chain. It is then only necessary to assign signals and instructions to the sequence chain for each step, when the control section is more or less complete.

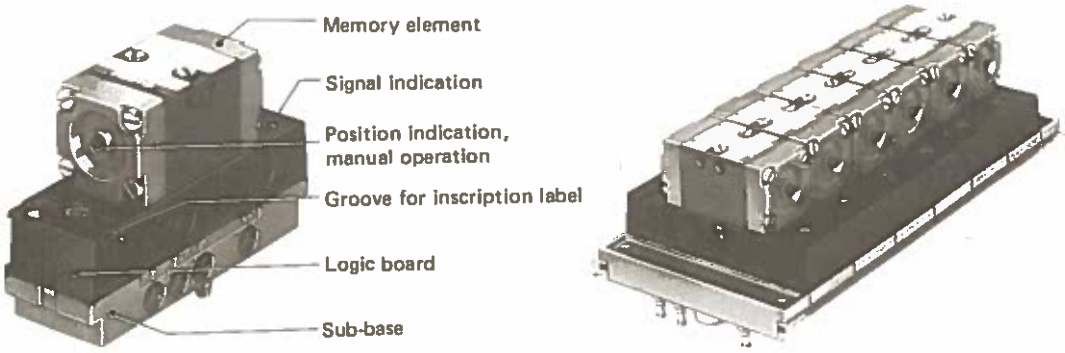


Fig 1 FESTO Pneumatics shift register module (left) and shift register chain (right). The preceding switching step of a shift register chain activates the next step. But the check-back signal of an executed movement (limit switch reached) is required before the switching step which has already been activated is set.

The following example illustrates how the Festo shift register method may be applied.

First Step: Specifying the cylinder movements.

1. Each cylinder is identified, *eg* alphabetically with A, B, C . . .
2. In our examples we are using double-acting cylinders. The control ports of the working valves are identified similarly to the cylinders.

   Cylinder advance: A+, B+, C+

   Cylinder return: A−, B−, C−
3. The positions of the piston rods are scanned in the front and rear end positions *via* limit switches. When retracted, the piston rods actuate the limit switches a_0, b_0, c_0. When extended, the piston rods actuate the limit switches a_1, b_1, c_1. (Fig 2).

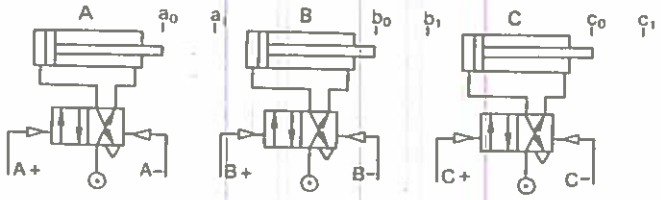


Fig 2

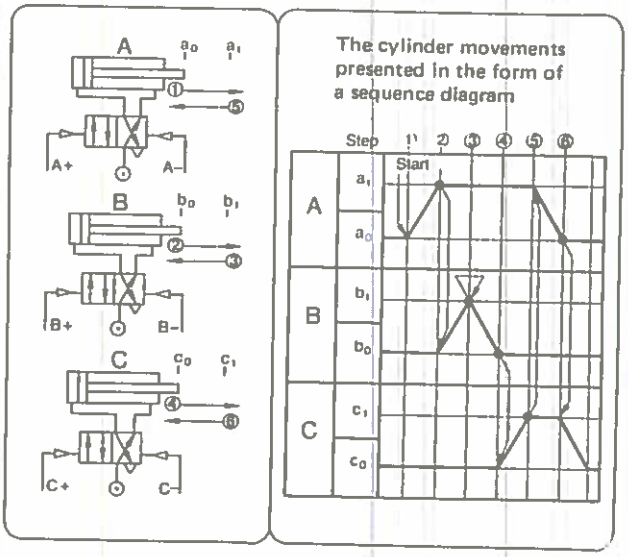
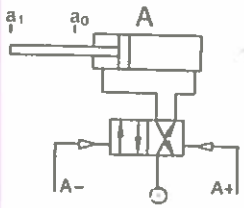


Fig 3



If the piston rod of cylinder A has extended, then limit switch a_1 is actuated.

If the piston rod of cylinder B has extended, then limit switch b_1 is actuated.

If the piston rod of cylinder B has retracted, then limit switch b_0 is actuated.

If the piston rod of cylinder C has extended, then limit switch c_1 is actuated.

If the piston rod of cylinder A has retracted, then limit switch a_0 is actuated.

If the piston rod of cylinder C has retracted, then limit switch c_0 is actuated.

Step	Movement	Limit switch
1	A+	a_1
2	B+	b_1
3	B−	b_0
4	C+	c_1
5	A−	a_0
6	C−	c_0

Second Step: Specifying the movement sequence (Fig 3)

The movement sequences of the individual cylinders can be shown in the form of a sequence diagram or in the form of a list.

1 Piston rod of cylinder A extends = A+
2 Piston rod of cylinder B extends = B+
3 Piston rod of cylinder B retracts = B–
4 Piston rod of cylinder C extends = C+
5 Piston rod of cylinder A retracts = A–
6 Piston rod of cylinder C retracts = C–

A list makes it clear which limit switches are needed for which steps – see Fig 3.

Fig 3 shows the control sequence and the assignment of the limit switches actuated by each movement.

Third Step: Assigning the sequence stages and forming a shift register chain (Fig 4).

Each step is assigned a shift register module.

Forming a sequence chain (Fig 5).

Assigning the instructions (cylinder movements) and the signals (check-back signals from the cylinders in the end positions) to the sequence stages.

Step-by-Step Sequence of a Shift Register Chain

Symbolic presentation of shift register chains in the form of a block diagram (Fig 6).

Step 1:

1.0 Start signal.

1.1 Stage (1) is set and an output instruction, *eg* A+ is triggered.

1.2 At the same time, stage (2) is primed and,

1.3 the last stage in the chain (in our case stage (6)) is reset.

Number of swtiching steps

Step	Movement	Signal
①	A+	a_1
②	B+	b_1
③	B–	b_0
④	C+	c_1
⑤	A–	a_0
⑥	C–	c_0

Number of sequence stages

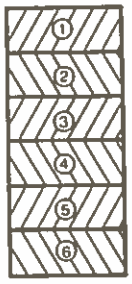


Fig 4

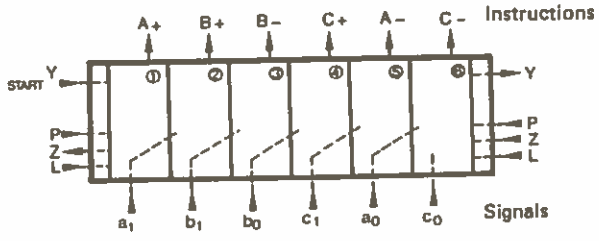


Fig 5

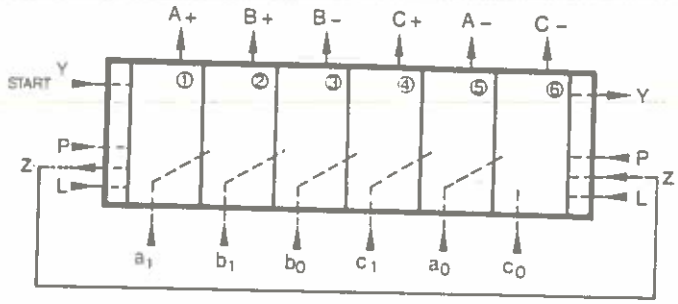


Fig 6

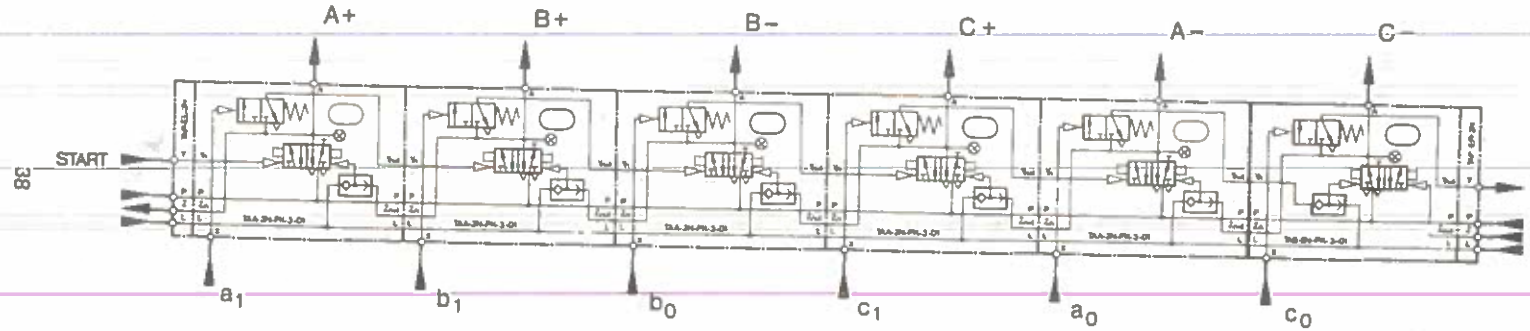


Fig 7

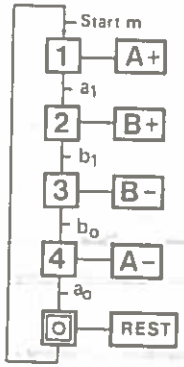


Fig 8 Grafcet diagram
Write down the desired sequence, assign a step module to each step and write the appropriate output signals and return signals each side of the assembly drawing.

Step 2:

2.0 When stage (1) receives the acknowledgment "instruction executed" from limit switch a_1, then:

2.1 Stage (2) is set and an output instruction *eg* B+ is triggered.

2.2. At the same time, stage (3) is primed and

2.3 stage (1) is reset (is now no longer required).

Step 3:

3.0 When stage (2) receives the acknowledgment 'instruction executed" from limit switch b_1, then:

3.1 Stage (3) is set and an output instruction *eg* B— is triggered.

3.2. At the same time, stage (4) is primed and

3.3. Stage (2) is reset (is now no longer required).

Step 4 and the following steps are similar — see Fig 7.

This sequence is ensured by an AND and OR function integrated in the shift register modules.

If a shift register chain gets stuck at any one of the stages, then the stage which is activated at that moment can be reset *via* a central correction signal at connection L. The central correction signal is applied to one of the two OR inputs.

A complete shift register chain is closed off by a left-hand and a right-hand end plate.

More elaborate automation schemes may be based on logic element controls and circuits, where *pneumatics* offers the advantage of an all-fluid system, using the same fluid medium throughout (compressed air). This can offer distinct advantages and savings over electronic controls and electro-hydraulic systems.

In this latter respect, fluidic controls are directly competitive with electronics with the limitation outlined above in the field of fluid power application, *pneumatics* is directly competitive with *hydraulics*. In the mechanization/automation field, pneumatics is generally superior to hydraulics in taking over direct from the manual operator. The superiority of hydraulics is largely limited to the higher output forces and power available, and to the greater rigidity of control movements directly available.

Logic Circuit Design

Circuit design for automated control based on logic functions in terms of hardware, means miniature control devices capable of performing AND, OR, NOT, INVERSION and MEMORY functions. The cost of such a system in pneumatic logic controls is directly related to its complexity. This does not necessarily apply with electronic controls.

An example of a simplified system is the Pneumatic Sequence, which consists of base-mounted modules, each module representing a step in a sequence. If there is a sequence with, for example, four steps, then a module is assigned to each step. When a 'start' signal is applied, module 1 becomes active and this sends a signal to start the first movement, the completion of which is sensed by a valve or some other sensing device. The signal from the sensing device is fed back to module 1, which then signals module 2 to operate. When module 2 becomes active it repeats the same process, whilst at the same time automatically resetting module 1. On completion of movement 2, module 2 activates module 3 which in turn resets module 2 and also maintains the reset on module 1. This is repeated step by step through the system until the last module, which resets itself on receiving its return signal, therefore resetting the whole system ready for another operation. Fig 8 shows the simple transition from writing down the sequence, to drawing the

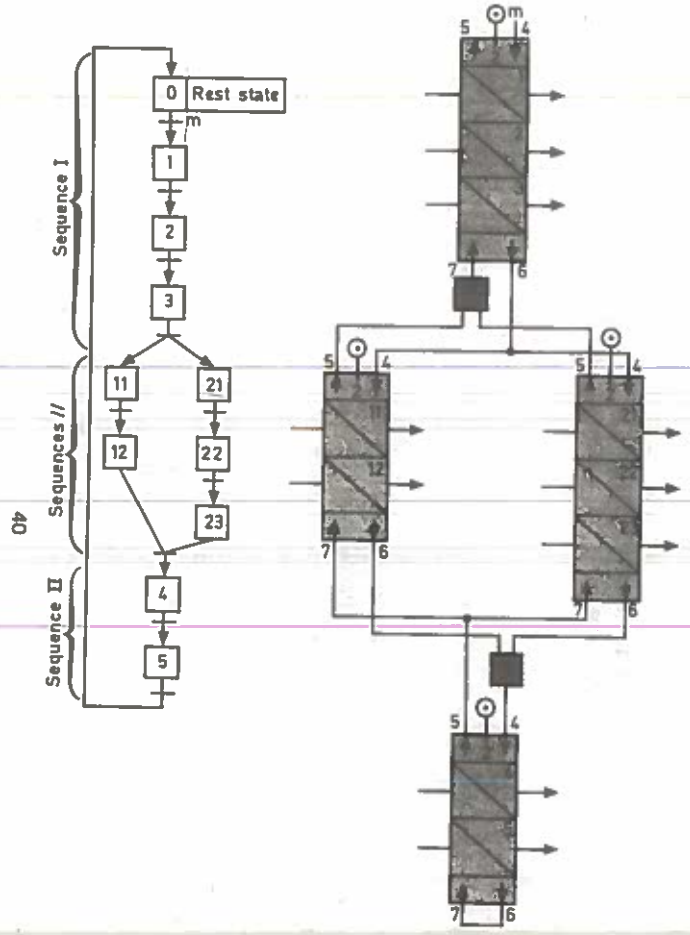


Fig 9 Transition from a divided path system to a circuit drawing.

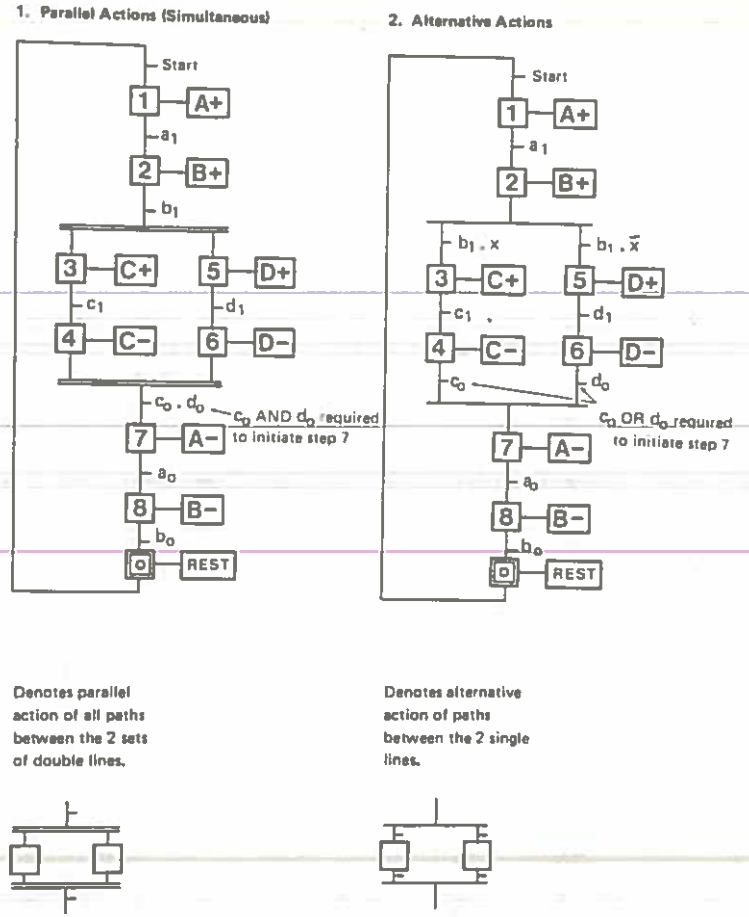


Fig 10 Divided paths illustrated by Grafcet.

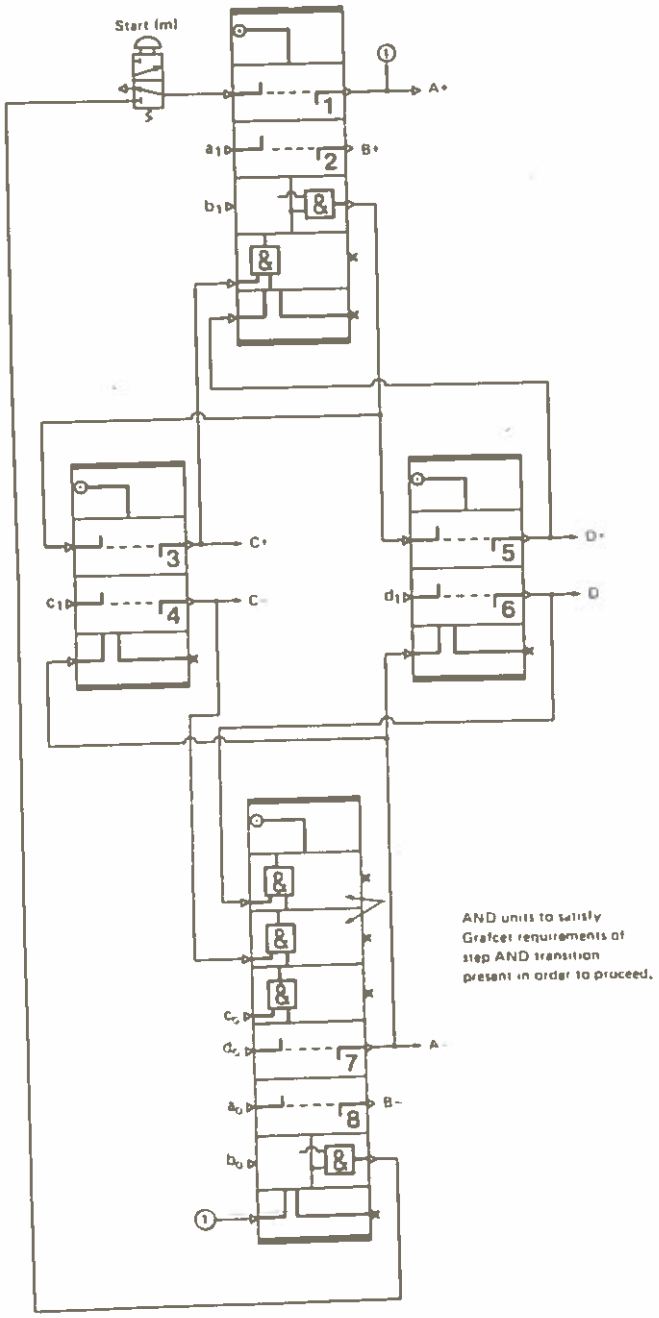


Fig 11 Parallel actions with polylog. (CompAir Maxam)

circuit and to the assembled unit. Therefore, if the sequence of operations required for the application can be determined, the circuit has virtually been designed. Each module is fitted with an indicator, so if a fault develops it is a simple matter to determine the area in which it has occurred.

Fig 9 shows how a complex divided path system becomes easy to design with this system, the circuit drawing and the final assembly following exactly the pattern of the written sequence. Should a time delay between movements be needed, then the timer unit is mounted on top of the relevant unit; similarly, if sensing devices other than a simple valve are required, then the associated amplifiers and relays can also be mounted on the modules in the same manner.

Several systems have, in fact, been developed to provide a simplified, easy-learn approach to pneumatic logic circuit design as an alternative to Boolean algebra, Karnaugh maps and Cascade techniques. These are related to specific modules and components, based on a particular *type* of logic (all solutions then being derived in that logic). Apart from basic logic units, the most commonly used device in such control systems is a timer for introducing time delays.

Further problems can arise on rotary and linear transfer machines where several stations, each following its own sequence, are operating simultaneously, calling for *divided path* systems. Again these can be solved by simplified techniques. Fig 10, for example shows a divided path problem represented graphically by Grafcet diagrams (a step-by-step function chart). Fig 11 shows the solution in the Polylog system.

See also chapter on *Pneumatic Logic*.

Safety

COMPRESSED AIR is an inherently safe form of energy in that it is free from fire hazard (such as is present with fuels and oils); and the potential hazard of electricity or high pressure steam. At the same time contained compressed air can expand explosively if suddenly released by failure of the vessel or component containing it.

Specific safety requirements of a compressed air system are therefore:

(i) All parts of the system should be protected against over-pressure. Such protective devices should be tamper-proof, where expedient.

(ii) All components must have a suitable working pressure rating relative to the system pressure employed.

(iii) Receivers should have an approved type and size of safety valve to relieve full input flow at tank design working pressure under all conditions (*eg* valves must not seize at freezing temperatures). Receivers and other types of pressure vessels should also be marked with hydraulic test pressure and date of test, design working pressure, capacity and standard or code to which they are manufactured.

(iv) All pneumatic systems should have a main airline shut off valve of the pressure-release type, capable of safely venting all working circuit pressure.

(v) Systems should be designed and constructed so that components are readily accessible for adjustment and/or renewing.

(vi) Systems should be designed on a fail-safe basis, so that in the event of failure, maximum safety to personnel is maintained, with minimum damage to equipment.

Individual consideration may need to be given to pneumatically operated devices and machines. These are relatively free of hazard and do not normally call for special provision of 'emergency stop', unless specifically required by relevant safety standards or the mode of working. This does not mean that pneumatic devices are automatically safe, for hazards are inevitably present with any working machine, particularly should it malfunction. It is essential, therefore, that a machine operator should be aware of the abnormal motions a machine may perform under certain circumstances and how to deal with them. This can apply particularly in the case of air cylinders.

The main hazard associated with an air cylinder is that its speed of action is governed by the exhaust opening and/or extenal load. Abnormal functioning is usually caused by a change in one of these 'control' parameters — usually the load. This can render the cylinder potentially dangerous as long as it is under supply pressure.

If, for example, the load is abnormal, the normal cylinder movement will be slowed, or even stopped. With the supply still connected pressure will build up on the piston, so that if the load is suddenly removed the piston will shoot forward at uncontrolled speed. This could cause injury to anyone in its path — the 'classic' subject in this case being an operator removing the blocking load with hands or feet. The correct action in such a case is both to shut off the supply *and* depressurize the cylinder before attempting to remove the blocking load.

Emergency stop controls are commonly provided on all complex machines and on larger cylinders and devices rated for medium and heavy duties. On light duty machines, 'emergency stop' is usually via the manual control valve. Where an emergency stop is provided this should be in a readily accessible position, separate or clearly distinguished from the other controls. It is also desirable that this control should have a locking or 'hold' action. Thus the usual form of 'emergency control' is a pressbutton. Once pressed, 'stop' should remain effective even when the button is released. Restarting can then be initiated by a separate control after the system or service has been cleared as safe, at the same time resetting the emergency stop button.

The mode of 'emergency stop' must also be considered in some detail, based on the characteristics and requirements of the device concerned. It may be desirable to stabilize the stop position, *eg* by locking the air in the cylinder on either side of the piston, or even more rigidly by means of a hydraulic check unit. In such cases the 'emergency stop' circuit would be designed accordingly. In other cases it may be necessary to avoid an abrupt stop as this could cause damage to the machine or introduce a further potential hazard. Here the desirable stop movement would be controlled, delayed or positioned (*eg* movement inched to a 'safe' position).

Reset characteristics can be equally important. The main requirement with air cylinders is that once reset has been initiated the possibility of excessive piston velocities must be avoided. One way in which this can be done is by using a three-way valve connected as a shuttle valve to a throttle valve in the supply line. The initial air supply on reset is thus throttled, enabling pressure to build up slowly in the cylinder. Once the required working pressure has been reached the three-way valve is automatically switched over to provide full flow — see Fig 1. An example of an emergency stop system with automatic resetting of main valves with slow replenishment of air is shown in Fig 2.

The simplest method of achieving an emergency stop is by using a three-position valve as the control, shutting off the supply and opening the system to the atmosphere. At the same time the associated electrical circuits can be tripped by the valve movement or broken by a pressure switch.

This basic arrangement can then be modified as necessary to incorporate other action, *eg* throttling, and the valve selected accordingly. Thus a four-position valve would normally be required to provide locking action or any *necessary* control. For example, it could be hazardous to evacuate a pneumatic clamp on emergency shutdown as this could cause the workpiece to be ejected violently. In other cases additional motions may be necessary to render the system safe on shutdown.

Systems in which loss of mains pressure can lead to hazard (*eg* incorporating machines with clamping components) can readily be protected by automatic emergency stop sensors, such as pressure switches. Fig 3 is a simple example using a non-return valve as a sensor to protect the clamping cylinder from loss of pressure. Once clamping pressure has been built up by the supply,

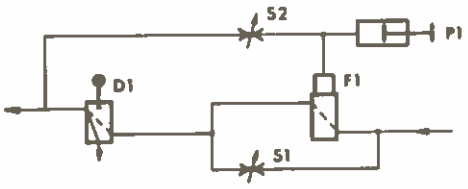


Fig 1

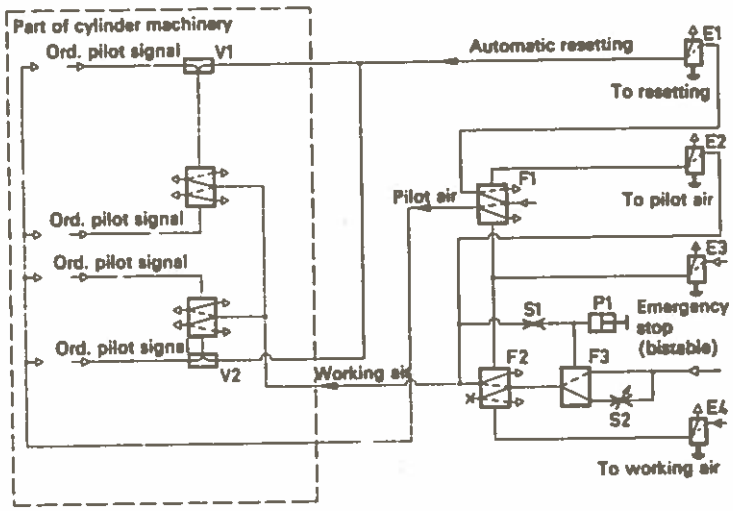


Fig 2 (from Mecman-Technique)

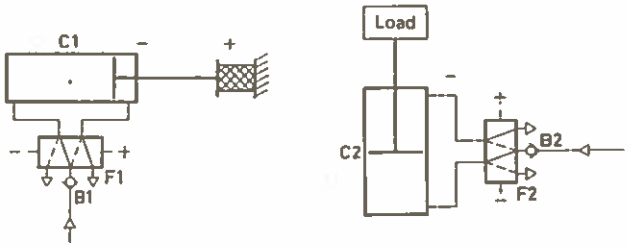


Fig 3 C1, C2 – cylinders.
B1,B2 – non-return valves.

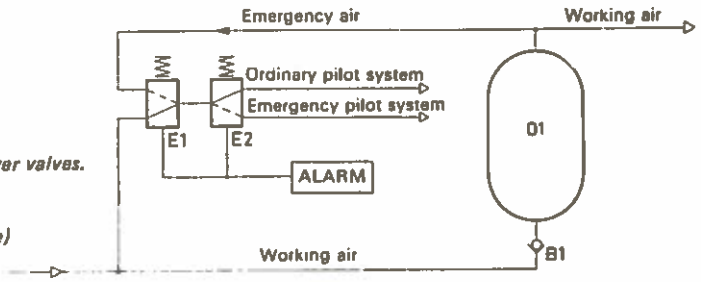


Fig 4 F1, F2 – changeover valves.
B1 – non-return valve.
01 – reservoir.
(from Mecman Technique)

any loss of pressure in the supply line will have no effect since there can be no backflow (and thus loss of pressure) through the non-return valve.

Fig 4 shows another type of circuit providing automatic stop in the event of the mains pressure falling below a certain level. At a given lower pressure, valves E1 and E2 are switched over. The reservoir now supplies working air to the service line(s), to move them to a 'safety' position controlled by the emergency pilot signal.

Safety, in other words, can be built into most pneumatic systems, at very little additional expense. The more complex the system or the heavier the mechanical output forces involved, the more desirable the need for 'safety switching', either manually or automatically. The primary aims are to remove any hazard from the system; and at the same time to maintain all necessary control over the movements or cycle involved. The first provides direct operator protection, and the latter can both provide further protection for the operator and safeguard the machine elements involved. Built-in safety can thus also provide economic benefits.

Examples of Fail-Safe Cylinder Circuits

The actual control may vary from simple open/closed condition to that of modulating control by a servo cylinder. Obviously, it must be decided at what level the pressure is classed as failed, and once the pressure has dropped to below the minimum which is essential to carry out the desired function, failure must be registered. Therefore it is very important to have an adjustable pressure switch, which can be set to switch at the desired failure pressure. Fig 5 shows a typical circuit.

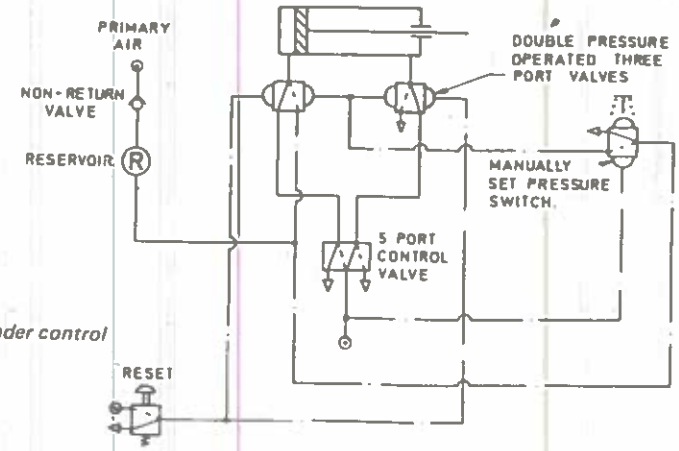


Fig 5 Double-acting cylinder control

Normally, in the healthy state the cylinder is controlled by the 5-port valve, via the two double pressure operated 3-port valves. In the event of the primary air failure, the pressure switch will sense and switch, reversing the two 3-port valves. The front end of the cylinder will now be vented and the reservoir, which has been charged during healthy conditions, charges the rear end of the cylinder, causing the cylinder to outstroke. The non-return valve is necessary to ensure that the reservoir pressure is not fed back into the air main in the event of primary air failure.

Obviously, the cylinder circuit can be designed to fail to the instroke position, if required, by merely changing over the valve connections. When the primary air supply is restored the reset valve

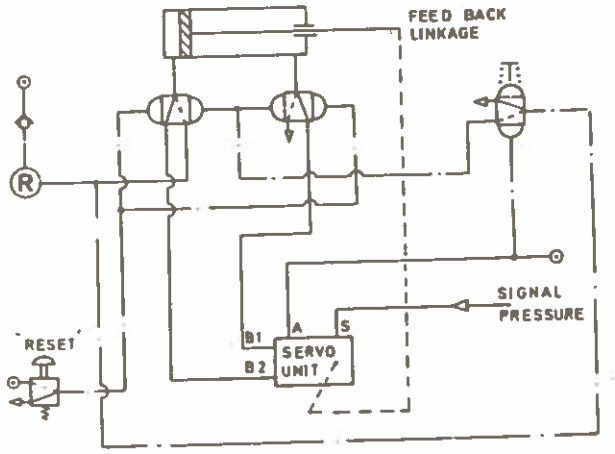


Fig 6 Servo cylinder control

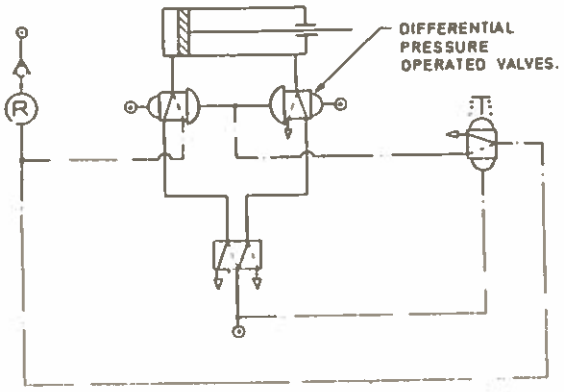


Fig 7

is operated to set the circuit to normal function. Fig 6 shows a similar arrangement but in this circuit a servo or positional cylinder is being used.

In some applications a reset valve may not be required and it would be more advantageous for the system to reset automatically when the primary air supply is restored. This arrangement is obtained by using differential operated 3-port valves as shown in Fig 7 and of course this also can be applied to the servo system.

The reservoir size is very important as it is necessary to ensure that when the cylinder has made its fail safe movement by utilizing the reservoir air, the pressure in the reservoir is still above the safe working pressure for the system. It can be seen that if the size of reservoir is calculated incorrectly the pressure could drop during a fail safe movement to below the safe working pressure of the system, and therefore the cylinder could stall in an unsafe position. It would therefore be useful to investigate the calculations necessary to arrive at the correct size of reservoir.

It should be noted that the double pressure operated valves and the reservoir should be mounted as near to the cylinder as possible, so avoiding losses in the pipe line and reducing the distance/time lag in the emergency system. The pilot line from the pressure switch should also be

of minimum length and 3/16 inch o.d. tube size is recommended. Obviously, the location of the pressure switch should be such that it cannot be tampered with.

Cylinders Failing to the Hold Position

On occasions it is required to hold the cylinder in the position occupied at the time of air failure. This requirement generally applies to servo cylinders which normally operate between the stroke extremities.

The solution of this problem is achieved without the use of a reservoir, as shown in Figs 8 and 9. Normally the two poppet valves are held open by a signal from the pressure switch. In the event of air failure the signal from the pressure switch is removed and consequently the two poppet valves close under spring pressure. The air in the cylinder is now trapped, resulting in the cylinder being pneumatically locked. This locked or held position is only relative to the terms of reference already outlined.

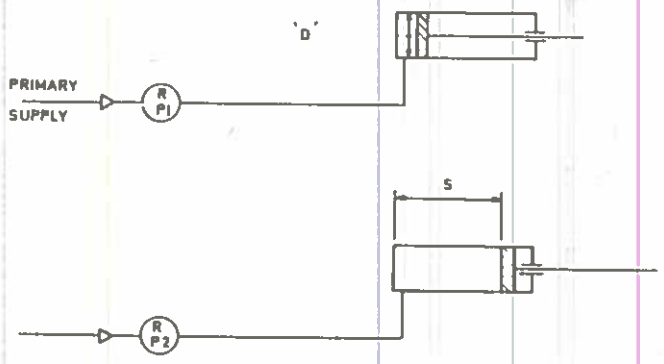


Fig 8

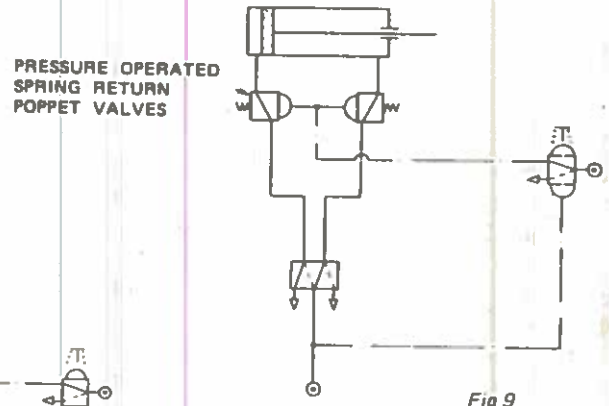


Fig 9

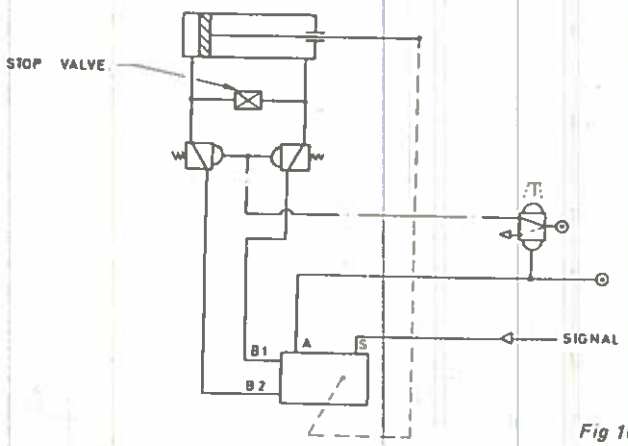


Fig 10

If it is found necessary to override this condition manually prior to resumption of primary supply, a stop valve can be added to the circuit as shown in Fig 10. This is normally closed, but by opening the stop valve during the lock condition the two ends of the cylinder are connected and therefore the pressures are balanced out and it is possible to move the cylinder manually.

Checking Safety Arrangements

For safety reasons it is very important that the fail safe arrangements are checked at regular intervals to ensure that they function satisfactorily. This can be provided for by the inclusion of an override valve in the line between the primary supply connection and the pressure switch (Fig 11).

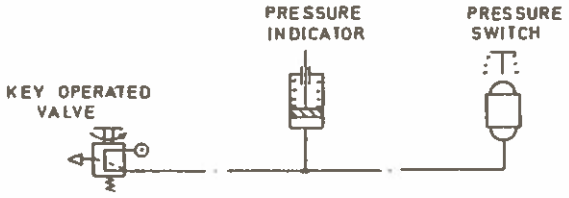
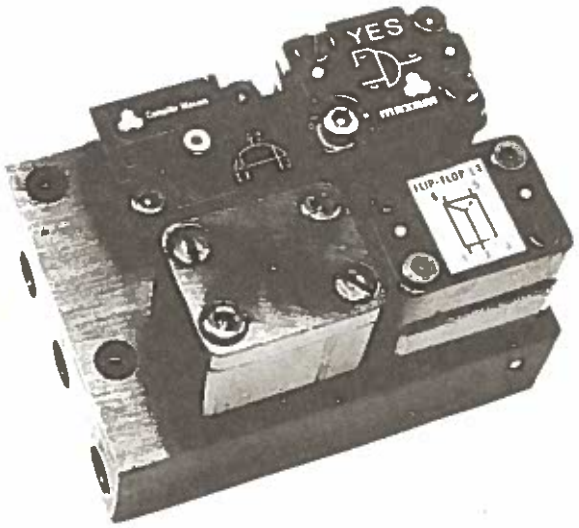


Fig 11

Inspection can now be carried out by suitable personnel, who, on turning the key in the valve, would stimulate the loss of primary supply pressure by putting the pilot line to atmosphere. An indicator could be fitted to supply a visual means of ensuring that after the test the primary supply has been restored.



'Bi-Manual' two handed safety control system provides a signal suitable to operate single-acting cylinders or pilot control valves to operate double-acting cylinders.

Cost Effectiveness

IN TERMS of specific energy requirement the cost of compressed air is somewhat higher than that of electricity. However, taken in isolation, direct costs of energy are only part of the overall question of cost-effectiveness, which must cover utilization (compressed air systems being suitable for some jobs where electricity is not; or in direct competition with electric motors in other cases, for example); capital costs and depreciation; overall efficiency; reliability; and operating costs (maintenance and labour). Cost-effectiveness aims at optimizing all parameters; and more particularly instituting savings when possible.

It is, for example, possible to achieve near 100% overall efficiency with a compressed air installation. This can be done by recovery of the compression heat generated in the production of compressed air.

When compressed air is used in a factory to perform work we count on having at the pneumatic machine an absolute pressure of 7 bar; that is, the overpressure above atmospheric pressure is 6 bar. The compressed air temperature is the same as in the factory, say 20°C. We now consider this compressed air as it is expanded back to the atmospheric pressure of 1 bar, having performed work in an air motor with no loss. In theory, one would obtain work corresponding to 29 kWh for each 1 000 m^3 of free air (at 1 bar).

During expansion, the temperature of the air will drop from 20°C to −105°C. The greatly chilled exhaust air must now take from the factory air an amount of heat corresponding to work performed in the air motor, 41 kWh. In simple terms, the compressed air, as it performs work under expansion, gives off internal energy and takes it back from the surrounding atmosphere after expansion. But in reality, a good air motor can only extract about 15 kWh in the form of mechanical work from 1 000 m^3 of free air, that is, less than half of the theoretical energy content of the compressed air.

Taking into account all losses in the airline delivery, compressor action and so on, the production of 1 000 m^3 of free air needs a compressor working to compress air to 8 bar absolute pressure. During compression, all the work developed by the prime mover is converted into heat, which must be led off by cooling in the compressor cooling system. Since losses in the prime mover — say an electric motor — and the compressor are also in the form of heat, we can simply say that the electrical energy consumed is converted into heat in the compression chamber and must be removed from it.

Small and medium sized compressors are air-cooled, and heat is usually taken off with the ventilation air. On larger types water cooling is used to the same end. Now if the waste heat can be utilized for heating nearby facilities very good overall economy is possible. Say in one year 500 000 m^3 of free air are produced so that the total energy represented would be 500 x 125 = 62 500 kWh (125 kWh are required to produce 1 000 m^3).

If we can use this thermal energy for, say, eight months of the year, then the amount of heat would be 42 000 kWh which is roughly what two normal sized houses need for heating and warm water during a year. Alternatively, it corresponds to the heat given out from 6 m^3 of fuel oil burned in a domestic furnace in a central heating system.

In the course of the year we obtain about 7 500 kWh worth of work from the 500 000 m^3. In all total utilization would be 42 000 + 7 500 = 49 500 kWh out of 62 500 kWh consumed in the production of the compressed air. So overall efficiency could be 80%.

The efficiency of water-cooled installations can be higher, as heated water can be used for such things as laundries or shower rooms during summer too. About 100 kWh may be sacrificed in producing 1 000 m^3 in the large compressor, so some 80 kWh can be recovered from the cooling water. Add to this the work performed by the compressed air and overall efficiency tops 95%. Thus it can be extremely cost-effective in many cases – particularly with larger installations, to utilize the waste heat generated in compressed air systems.

At the other end of the scale *leakage* is one of the greatest air consumers and can account for 5–10% of installed compressor capacity. Taking a process industry as an example, with an installed capacity of 2 m^3/sec and leakage estimated at 5%, the consumption including leakage amounts to 1.5 m^3/sec during two day shifts and 0.5 m^3/sec during the night shift. A year's consumption amounts to

$$(1.5 \times 3\,600 \times 4\,000) + (0.5 \times 3\,600 \times 4\,000) = 29\ m^3 \times 10^6.$$

Total leakage during the same period amounts to

$$0.1 \times 3\,600 \times 8\,000 = 2.9\ m^3 \times 10^6 \text{ or } 10\% \text{ of total consumption.}$$

In other words, twice the estimated leakage. For this 'moderate' leakage, energy costs will be 290 MWh on the basis that 1 kWh produces on average 10 m^3 of free air at 7 bar.

Regular leakage measurements will keep leakage within reasonable limits. By measuring when production is at a standstill and when a compressor with known air flow output runs loaded in relation to the total running time needed to keep operating pressure at a normal value, one can calculate the true leakage. It is possible to localize leaks by closing off different sections during measurement. Regular checks should be made on local connections to air tools and equipment.

Another important factor governing leakage is maintenance of air tools, actuating cylinders and valves. Air consumption increases in worn equipment at the same time as output falls off. Correct operating pressure is also important. Increasing air pressure for temporary increases in output only accelerates wear rates, which ultimately increase air consumption. Reduced pressure means great pressure drops in airlines, and output of power from an air tool drops quicker than the rate of pressure fall. So again, for work out at lower pressure, consumption increases.

It is not merely a question of taking conservation steps at the compressor source, discipline in maintenance must apply throughout the system and air tooling if full conservation is to be realized.

Specific Energy Requirement

Specific energy requirement is defined as the (input) energy spent on compression of unit volume of air to a specified pressure ratio. For compressors driven by electric motors this is expressed

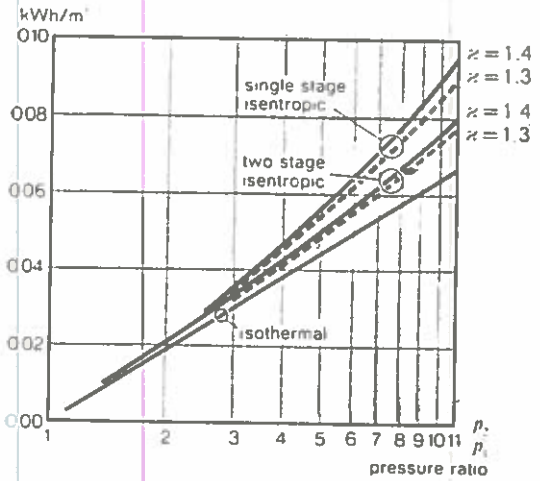


Fig 1 Specific energy requirements for compression of air from atmospheric pressure. (Atlas Copco).

in units of joules per litre (j/l) or kilowatt hours per cubic meter (kWh/m^3) — see Fig 1. For compressors driven by internal combustion engines, the unit is grams of fuel per cubic metre (g/m^3).

Energy costs are usually the largest item, emphasizing the importance of reducing power consumption wherever possible — *eg* by optimized unloading of the compressor; and also suitable choice of compressor size. The latter can be a significant parameter in the case of an installation where demand is likely to increase, since efficiency tends to increase, and thus specific power requirements decrease, with increasing compressor size.

Compressor Power Input Required

The input power required to generate compressed air at the normal air line pressure of 7 bar (100 lb/in^2) is of the order of 15 kW per 47 l/min (100 ft^3/min). This will vary somewhat with the type of compressor and its overall efficiency, and also with the actual pressure generated — see Table I.

TABLE I — APPROXIMATE POWER REQUIRED TO GENERATE COMPRESSED AIR — POWER IN kW

FAD	PRESSURE — bar				
m^3/min	4	5	6	7 (100 lb/in^2)	8.75 (125 lb/in^2)
1	4.1	4.5	4.9	5.3	5.8
2	8.3	9.0	9.8	10.6	11.6
3	12.3	13.5	14.7	15.9	17.4
4	16.4	18.0	19.6	21.2	23.2
5	20.5	22.5	24.5	26.5	29.0
6	29.6	27.0	29.4	31.8	34.5
7	28.7	31.5	34.3	37.1	40.6
8	32.8	36.0	39.2	42.4	46.4
9	36.9	40.5	44.1	47.7	52.2
10	41.0	45.1	48.8	53.0	58.3

The true cost of generating this input horsepower depends on the efficiency of the driver. Electric motors are usually the most economical type of driver, since they are capable of achieving efficiencies in excess of 90%, offsetting what is apparently a high 'fuel' cost.

A further allowance may have to be made for drive losses, where the driver cannot be directly coupled to the compressor. Thus i/c engines are relatively low speed machines and are suitable for direct coupling to lower speed compressors (*eg* reciprocating compressors). They would need to be geared up to drive high speed rotary compressors, or such compressors could suffer a loss of efficiency with direct drive. In the case of electric motors, driver speeds are again limited to a maximum of 3000 rev/min with synchronous motors on 50 cycle supply.

An important point is that where demand is high and fairly continuous, efficiency figures can be very significant. A relatively small difference in compressor, driver or drive efficiency can represent a substantial difference in compressed air costs over a period of a year or more. Choice of compressor type can, therefore, be significant – see chapter on *Compressor Selection.*

Compressor Regulation

Either constant or variable speed control may be used to advantage, where demand is fairly continuous, for this will usually produce a minimum overall cost for the compressed air consumed. Variable speed control is rather better than constant speed control in this respect, although demanding a more complex control arrangement.

Stop and start control is simpler and generally more economical on large systems subject to intermittent demand. Here the compressor charges the receiver in the normal way, switching off when a predetermined pressure is reached and remaining idle until the receiver pressure has fallen to a predetermined low level. The compressor is then automatically restarted and continues running until maximum charge has been reached once more.

The economies of stop and start control depend largely on the number of stops and starts per hour, which is related to the receiver size and demand. The greater the number of stops and starts per hour, the higher the wear and tear on the compressor and driver, and the control gear. For any receiver size, the maximum number of stops and starts per hour will be realized when the air consumption figure is one half of the FAD (Free Air Delivered) of the compressor. To hold the number of stops and starts to a reasonable level, a generous receiver size is usually adopted, *eg* typically about twice that required with constant speed regulation. See also chapters on *Compressor Controls* and *Air Receivers.*

Air Mains

Distribution of the compressed air is by pipe, or air main, from a receiver following the compressor which may be either a single line terminating in a dead end, or a ring main. In both cases the main is normally pitched in the direction of air flow so that condensed water will be carried along away from the receiver end, to collect in traps or drains installed at suitable points. This is an essential feature of economical operation as maximum efficiency and maximum appliance life is achieved only by the use of clean and preferably dry compressed air.

One of the most important control parameters in the overall economics of a compressed air system is optimum line sizing. Delivery of compressed air through the air mains, branch lines, and to industrial tools and appliances, immediately results in a pressure drop, and thus energy loss. These losses are minimized by optimum pipe sizing based either on a specific design pressure drop, or an 'acceptable' pressure drop (usually of the order of 5 to 10%). It is important to remember in this connection that pressure drop (and thus line sizing parameters) are dependent on air velocity and thus also governed by demand.

The subject of line sizing is detailed in the chapters on *Compressed Air Mains* and *Branch Lines.*

TABLE II – AIR LOSS AND POWER WASTAGE BY AIR LEAKS†

Line Pressure		Leakage in ft^3/min				Approximate Power Wastage			
hour	lb/in^2	1/16 in	1/8 in	1/4 in	1/2 in			in kW/hr	
		3.5	1.45	59	235	0.65	2.6	7.8	25.4*
7.0	100	6.5	26	104	415	1.25	5.2	18.5*	59*
10.0	150	9.5	37	150	600	1.6	6.4	26*	104*

* = two stage compression

†B.C Regan (Spirax Sarco Ltd).

Air Leaks

Air leaks are not hazardous, but they can be extremely wasteful of power. In a typical system as much as 10% of the power requirements at any period can be made up of air leaking to waste. Table II gives some idea of the leakage possible through various size holes.

It is a simple exercise to determine the magnitude of the leaks in a compressed air system at some time during the day, perhaps during the lunch break when the plant has no programmed air requirements, by checking what percentage of time the compressor is on-load. This on-load time and power is being used exclusively to maintain leaks. A little extra thought and money on ensuring a good initial installation will result in a leak-free, less expensive compressed air plant. Apart from savings on running costs, a leak-free system will defer the need to purchase additional expensive and probably unnecessary compressor plant.

Actuator Efficiencies

Conversion efficiency achieved by an actuator powered by compressed air, ranges from about 90% down to 65% or lower in the case of linear actuators, depending primarily on the speed of operation. Conversion efficiency is also affected by the compression ratio, or actual air pressure at the actuator, normally reaching a maximum at pressures of 7–10 bar (100–150 lb/in^2) with conventional designs of actuators. Moderate pressures of this order reduce pumping costs and also the stresses on lines and components subject to internal pressure.

Actuators

In general a pneumatic system is the least expensive of any type where linear motions are required, provided the output force required is within the practical range of air cylinders. Compressed air is also unique in its suitability for powering certain types of tools such as percussion tools for road-breaking and rock drilling, *etc*, and is directly competitive with electric motors for powering small workshop tools.

Linear actuators powered by compressed air are inherently fast acting, but unrestricted high speeds can bring about high pressure drop at the inlet port, with consequent loss of effective pressure applied to the piston. Rarely, however, is a linear actuator called upon to work at its fastest speed, and efficiency can be improved by restricting the speed attained via simple throttling controls. Thus an 80% conversion efficiency is readily achieved in the majority of applications, with static thrusts or break-out forces equal to the full theoretical performance. Whilst this working efficiency is lower than that obtained with hydraulic systems, it is still high, and the

overall economics of compressed air as a working or 'power' fluid is often better than that of hydraulics, or indeed any alternative system. Further advantages are often apparent with the convenience and simplicity of installation offered by compressed air systems.

The displacement demand calculated directly from cylinder geometry represents the volume of compressed air consumed per stroke or cycle. The equivalent consumption in terms of Free Air Delivered (FAD) is given by multiplying the displacement by the compression ratio (r), *ie*

$$r = \frac{P_2 + P_1}{P_1}$$

where

P_2 = compressed air gauge pressure

P_1 = atmospheric pressure

Compressor capacity is normally given in m^3/minute or ft^3/min FAD (Free Air Delivered). If compressor capacity is rated in terms of free air *displaced,* this displacement figure must be multiplied by the compressor efficiency to obtain the true FAD figure.

Air Tools

In the case of pneumatic tool and other appliances, air consumption figures are normally based on empirical data. These are normally maximum demand figures, consistent with continuous operation. Actual demand will depend on the *time factor* and *work factor* involved.

The *time factor* is concerned directly with the time the appliance is in use. Thus tools and appliances are normally used intermittently rather than continuously, when a work factor can be based on observation over a suitable period of use. In the case of linear actuators or devices allied to a work cycle, the time factor can be established more accurately by time and motion studies, or calculated directly from the cycle time, if appropriate.

The work factor is related to the actual period of time during use when the appliance is consuming air. This can vary from 1.0 where the appliance is continuously driven by air when being used, to appreciably less where the demand is other than constant. For example, actual demand may vary according to the work involved, particularly if the load changes, or if the speed of operation is varied by a throttle control.

The product of the time factor and work factor represents the true load factor, or true demand from the compressor, from a single tool or device. The total demand is thus the sum of all the individual air consumption figures multiplied by their respective load factors. This will normally be considerably less than the sum of all the maximum air consumption figures for all the appliances connected to a system. Load factors can be expressed as a percentage or decimal fraction — see Table III. The true demand is then represented by the sum of the individual maximum demands of all the appliances connected to the system, when each is multiplied by its respective load factor.

The reduced air consumption in each case is derived directly by multiplying the maximum air consumption figure (column 2) by the load factor. The final total arrived at is 35.4% of the demand, as given by summing all the individual maximum air consumption figures. Provided the time and work factors have been correctly established, this lower figure would apparently meet all the requirements.

In practice this is not necessarily so, since the study does not analyze the instantaneous demand at any particular time. It merely assumes that the time factors are employed simultaneously, whereas they are not. For any time factor less than unity, there will be periods when the actual demand is equivalent to a time factor of 1.0, *ie* when the appliance is consuming air at maximum

TABLE III – LOAD FACTORS

Work Factor	TIME FACTOR 0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
0.1	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09	0.10
0.2	0.02	0.04	0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20
0.3	0.03	0.06	0.09	0.12	0.15	0.18	0.21	0.24	0.27	0.30
0.4	0.04	0.08	0.12	0.16	0.20	0.24	0.28	0.32	0.36	0.40
0.5	0.05	0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.45	0.50
0.6	0.06	0.12	0.18	0.24	0.30	0.36	0.42	0.48	0.54	0.60
0.7	0.07	0.14	0.21	0.28	0.35	0.42	0.49	0.56	0.63	0.70
0.8	0.08	0.16	0.24	0.32	0.40	0.48	0.56	0.64	0.72	0.80
0.9	0.09	0.18	0.27	0.36	0.45	0.54	0.63	0.72	0.81	0.90
1.0	0.10	0.20	0.30	0.40	0.50	0.60	0.70	0.80	0.90	1.00

rate. If two or more of these periods coincide, as they are very likely to, then the true total momentary air consumption figure will be higher than the reduced figure, although the *total* demand over a period will equal that of the reduced figure.

In general, it is usually satisfactory to estimate on the basis of total reduced air consumption and allow a generous size of receiver to take care of momentary peak demands in more detail and estimate mains sizes (the latter particularly) rather than design to marginal values.

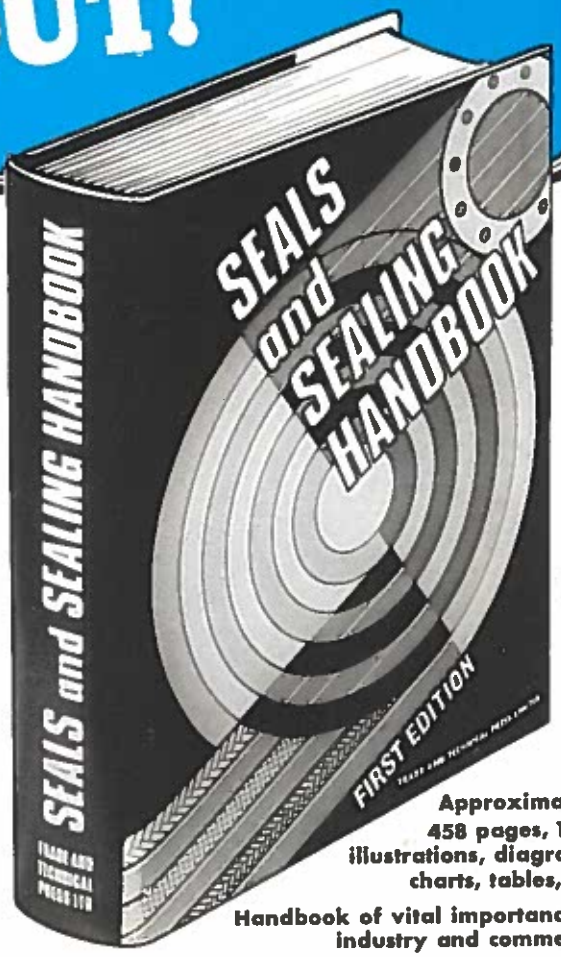
Actual demand will also vary with pressure and ambient air temperature (and compressor delivery will also vary with air intake temperature), but these factors can generally be ignored if the compressor capacity and receiver volume required is generously estimated in the first place. Some readjustment of capacity requirements (and loss of compressor capacity) may have to be made for working at altitude, however.

Stocking of spares can represent a major capital cost item, with low utilization where a large number of air tools are employed. Ideally stocks of spares should be carried covering the normal regular replacement items likely to be required at scheduled maintenance intervals. Stocking an extensive range of spares is, however, not always an economic proportion, even though a large number of individual tools may be involved. This is because a considerable amount of capital may be tied up in spares which are seldom required. On the other hand, lack of a particular spare may put a wanted tool out of action until the necessary replacement part has been ordered and delivered.

An alternative solution, which can prove most effective, is to carry only a minimum stock of necessary spares – *ie* those normally expected to be required at short notice during scheduled maintenance – together with a spare stock of complete tools. A faulty tool can then be replaced immediately from the surplus stock should it fail before its scheduled maintenance period; or, if a necessary replacement part is not available from stock at the scheduled maintenance period, the tool can be replaced and the part ordered. When delivered, the original tool is fully serviced and becomes part of the stand-by stock.

Such a system normally demands a minimum of 10% of the number of tools in regular, continuous use as stand-by stock, but may reduce the value of the spares stock which would otherwise be necessary by a similar capital amount. Having the bulk of the 'spares capital' in tools will show further benefits in minimizing shut-down time, particularly when dealing with emergency breakdowns.

JUST OUT!



EDITORIAL CONTENTS INCLUDE:—

SECTION 1 FUNDAMENTALS AND PRINCIPLES
Definitions — Mechanics of Sealing — Seal Performance (Friction, Leakage and Wear) — Environmental Health and Safety.

SECTION 2A STATIC SEALS
Gaskets — Sealing Rings and Washers — Liquid Sealants.

SECTION 2B DYNAMIC SEALS
Compression Packings — Automatic Packings — O-Rings — Simple and Composite Ring Seals — Lip Seals — Other Proprietary Seals — Wipers, Scrapers and Boots.

SECTION 2C MECHANICAL FACE SEALS
Design and Application — Different Types — Face Seal Materials.

SECTION 2D SPECIAL SEAL TYPES
Bellows Seals — Diaphragm Seals — Piston Rings — Bushing Seals — Labyrinth Seals — Visco Seals — Ferrofluidic Seals — Inflated Seals — Miscellaneous Seals.

SECTION 3 GENERAL APPLICATIONS
Rotary Shaft Seals — Oil Seals — Reciprocating (Rod) Seals — Gland Design — Seal Installation, Care and Maintenance.

SECTION 4A SEALS FOR SPECIFIC APPLICATIONS
Hydraulic Seals — Pneumatic Seals — Marine Packings and Seals — Controlled Leakage Seals.

SECTION 4B SEALS FOR SPECIAL APPLICATIONS
High Pressure Seals — Vacuum Seals — High Temperature Seals — Low Temperature Seals — High Speed Seals — Large Diameter Seals.

SECTION 5 MATERIALS
Terminology — Elastomers (Materials and Properties) — Plastics — Metals — Lubrication.

SECTION 6 DATA SECTION
Standards — Seal Selection Guides — Glossary.

SECTION 7 APPENDIX
Important Indépendent Organisations.

SECTION 8 BUYERS' GUIDE
Trade Names Index and Alphabetical List of Manufacturers and Distributors Names, Addresses, Telephone and Telex Numbers.

TRADE AND TECHNICAL PRESS LTD
CROWN HOUSE MORDEN
SURREY ENGLAND

Approximately 458 pages, 1500 illustrations, diagrams, charts, tables, etc.

Handbook of vital importance to industry and commerce.

Hard case, hand bound, gold blocked with dust jacket and bookmark.

ORDER YOUR COPY

NOW

Atlas Copco

We're getting Henry's favourite lady off her bottom.

The lady is a wreck.

She's Henry the Eighth's flagship, the Mary Rose. And she's been sitting patiently at the bottom of the Solent where she sank in 1545.

With so much historical value from stem to stern, a project was launched to raise her and take her to a new home on land.

Today, progress looks promising. But owing to the lady's delicate condition and the size of the operation it's had its problems.

Fortunately, companies like Atlas Copco have been able to lend their support. Having one of the world's leading ranges of compressors, air tools and ancillary equipment for the mining, construction and engineering industries we've been able to supply the technology essential for this type of salvaging.

For example, over 500 lorry loads of silt have been removed from the Mary Rose. And all this has been done by just two compressors, working a continuous 12½ hour shift, day after day through the diving season, without a single hitch. So it would be no surprise to find Mary Rose looks upon Atlas Copco with favour.

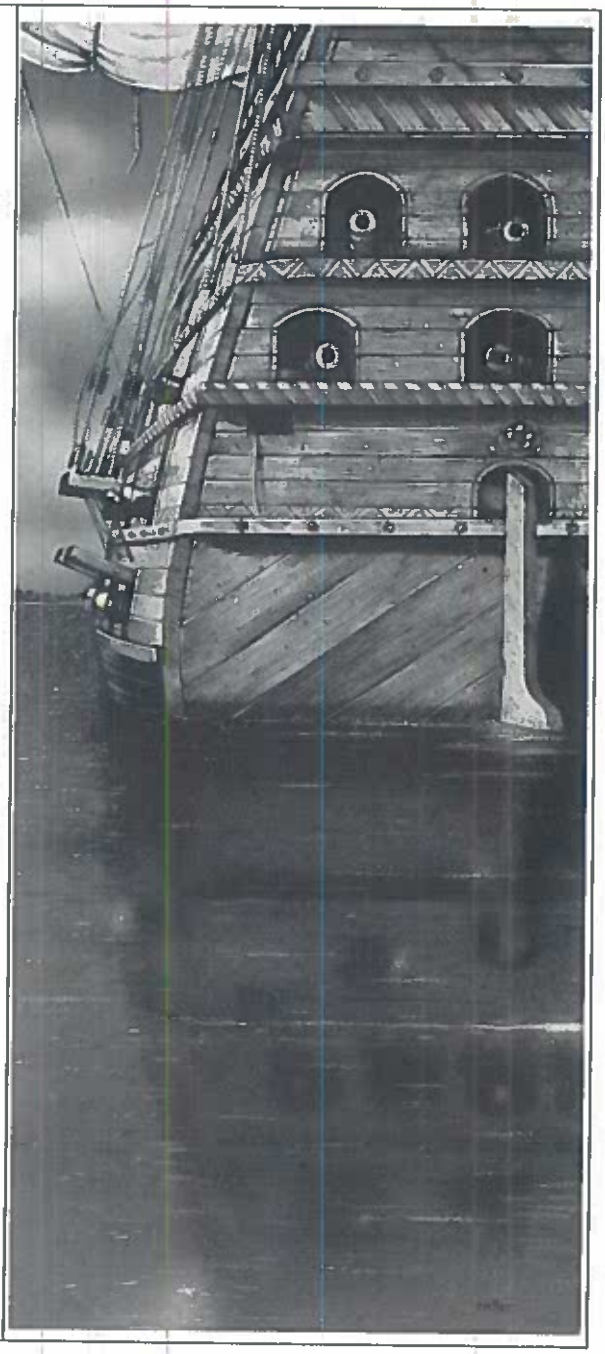
After all, she's been a lady in waiting long enough.

Contact us today for more interesting facts about the Atlas Copco range and our distributor network.

Chances are, if you've got a problem we'll soon get to the bottom of it.

Atlas Copco.
Profit from our experience.

Atlas Copco (Great Britain) Limited, P.O. Box 79, Swallowdale Lane, Hemel Hempstead, Herts. HP2 7HA
Tel: (0442) 61201



SECTION 2a

Compressors

Compressors

COMPRESSORS FALL broadly into two categories, *displacement* compressors and *dynamic* compressors. Displacement compressors induct and displace successive volumes of air (or gas) and work on the principle of increasing the pressure of a definite volume of air (or gas) by reducing that volume. Dynamic compressors produce a static pressure rise by converting kinetic energy into pressure energy during the passage of air (or gas) through the machine. Mechanically, the difference is that displacement compressors incorporate a moving member or members, such as a piston, to produce displacement. The working element in a dynamic compressors is an impeller.

Displacement compressors embrace both reciprocating and rotary types. Reciprocating compressors are the oldest and probably still the most common type of positive displacement compressor. Rotary compressors include screw, vane, lobe rotor and liquid ring types. During the last decade the oil-flooded screw compressor has come to the fore as a major type, largely because of the development of more efficient rotor profiles.

Dynamic compressors include radial flow, axial flow, and to a lesser extent mixed flow machines. The rotor form adopted is governed by capacity requirements, axial flow machines offering the largest capacities. It is a general characteristic of dynamic compressors that capacity varies with the working pressure – *eg* see Fig 1 for comparison with positive duplicate types.

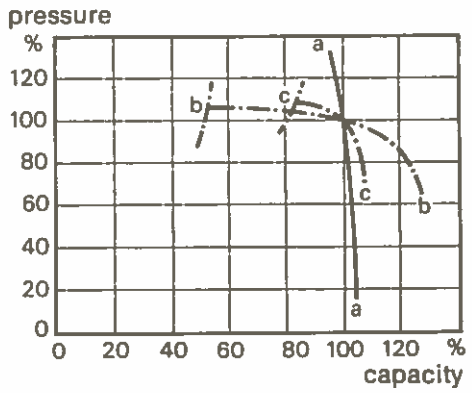


Fig 1 Constant speed compressor characteristics (a) positive displacement compressor, (b) centrifugal compressor, (c) axial compressor. (Atlas Copco).

Reciprocating Compressors

Reciprocating compressors are suitable for operating over a very wide range of speeds, with a practical maximum of about 10 bar (150 lb/in^2) delivery pressure from a single-stage unit; and up to 70 bar (1000 lb/in^2) from a two-stage machine. Multi-stage reciprocating compressors may be built for special purposes capable of supply delivery pressures up to or in excess of 700 bar (10000 lb/in^2). By carefully selecting the number of stages the designer can also produce a machine which approaches the ideal or isothermal compression curve more closely than any other type, with the possible exception of the very large volume axial flow compressors.

Reciprocating compressors are also available in both lubricated and non-lubricated designs. Non-lubricated compressors normally have piston rings together with bearing rings and skirts of filled PTFE.

In the case of larger compressors operating with closed cycle cooling, cooling is accomplished by passing the water through a mechanical cooler or a cooling tower. Again temperature control can prove advantageous, particularly to avoid too low a temperature being developed in the second stage cylinder.

In the double-acting compressor the space on the other side of the piston is also enclosed and both sides are used for compression, giving two compression strokes for each revolution of the crankshaft. Alternatively, one side of the piston may be used for one stage and the other side for another stage in a two-stage compressor. Individual cylinders may also be used for multi-stage compression, disposed in a number of arrangements.

The greatest number of reciprocating compressors are single-stage or two-stage, ranging from fractional horsepower units up to very large machines with input power requirements of the order of 2250 kW (3000 horsepower). The smaller compressors are usually single-stage of the single or vee-twin layout with air cooled cylinders and are powered by electric motors.

Intermediate sizes comprise a whole variety of different configurations. Thus vertical compressors may comprise one or more cylinders in-line, vee, 'W' and 'H' arrangements for multi-cylinder units, and also 'L' configuration which has both vertical and horizontal cylinders disposed around a common crankshaft.

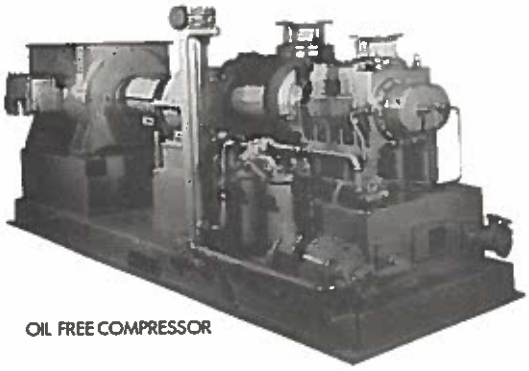
The 'angle' arrangement of cylinders offers certain specific advantages, notably reduced bulk and weight and superior machine balance (since all the primary forces can be accurately balanced). The 'L' configuration is advantageous for larger machines with the low pressure cylinder vertical and the high pressure cylinder horizontal, facilitating installation, assembly, maintenance and dismantling.

Larger reciprocating compressors are commonly horizontal double-acting, tandem, duplex, *etc.* Basic two-stage layouts are shown in Fig 2. Variations include transferring the sidethrust at the piston end of the connecting rod to the cylinder itself ('trunk' type); or relieving the piston of the sidethrust by means of a crosshead. When the space below the piston of a double-acting cylinder is used for pumping air, it is virtually essential to use a crosshead to convert the rotary action of the crankshaft into reciprocating motion in order to obtain a satisfactory seal of the piston rod where it passes through the compression chamber underneath the piston. The crosshead ensures that all sidethrust from the crankshaft is taken up in the crosshead guide and not transferred to the piston and cylinder. Single-acting reciprocating compressors are normally of the trunk-piston type, whilst differential-piston double-acting compressors can be of either type — see also Fig 3.

Differential or stepped pistons may also be used as shown in Fig 4. Here one stage of compression takes place in the annular space between the shoulder of the piston and the corresponding shoulder in the cylinder.

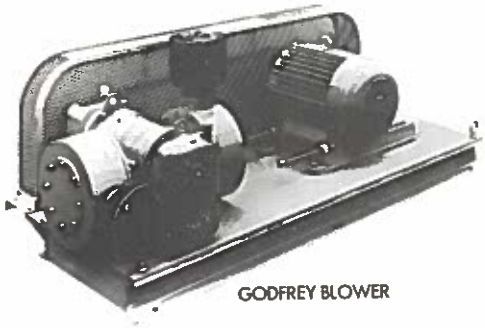
HOWDEN

Oil Free Screw Compressors
Howdenair Packaged Screw Compressors
Godfrey Blowers



OIL FREE COMPRESSOR

HOWDENAIR PACKAGED
INDUSTRIAL COMPRESSOR



GODFREY BLOWER

Oil Free Compressors
500 – 10,000 cfm (14 – 283 m³/min)
15 – 190 psig (1 – 13.1 bars g.)

Howdenair Packaged Industrial Compressors
65 – 1000 cfm (1·84 – 28·3 m³/min)
80 – 150 psig (4 – 10.3 bars g.)

Godfrey Blowers
30 – 1200 cfm (0·85 – 34 m³/min)
up to 15 psig (up to 1·03 bar g.)

Dependable air power –
for over 30 years!

Sales Office
Weybridge Industrial Estate
Weybridge, Surrey, KT15 2GR.
Tel. Weybridge 56631

Howden Compressors Ltd

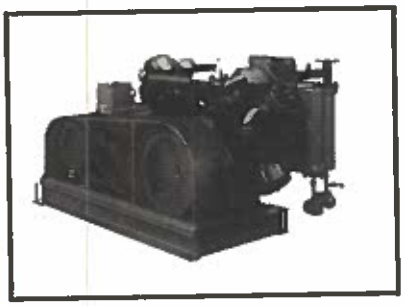
Glasgow G52 1BE
Tel. 041-882 3346
Telex 778711

Williams & James (Engineers) Limited
Chequers Bridge, Gloucester, GL1 4LL
Tel: Gloucester (0452) 28431 Telex: 43233

Williams and James have been engaged in high reliability engineering for over sixty years. During that time our major effort has been in the development of oil-free air and gas compressors.
We specialise in giving high quality air with exceptional machine reliability. These parameters give the W & J range of compressors their reputation for quality throughout the world, and especially in the control and instrumentation market, which has such stringent requirements. Our range of oil-free compressors has been developed to meet these requirements and are considered the best available.

At Williams and James we believe that we have an obligation to advise our customers not only on plant selection to ensure the correct unit is used for the job, but also to supply the necessary operational know-how and spares support for the installation.

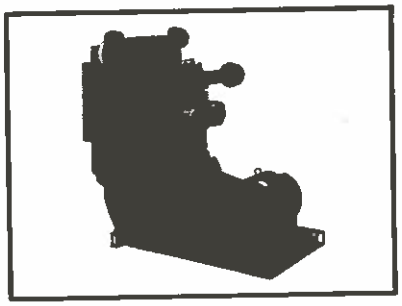
In order to extend this service during the compressor's working life a full maintenance and service contract is offered to cater for any eventuality.



Series 695
Outputs up to 200 cfm
Pressures up to 150 psi

The series 695 compressor is designed to provide top quality air for instrumentation and control, breathing air and other demanding applications. Being air cooled, it has found many applications in arid, cold and marine environments because it does not involve the extra installation and maintenance costs of water-cooled machines.

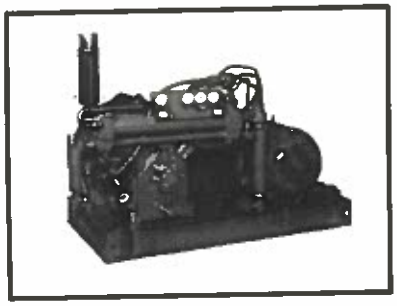
The 695 is a two-stage, two cylinder compressor incorporating piston rings running in dry cylinder bores. Integral air-blast inter and after coolers are incorporated. Crankcase motion work is force lubricated, and dynamic balancing with vee construction gives smooth, vibration-free running and high reliability.



Series 190
Outputs up to 120 cfm
Pressures up to 125 psi

Series 190 are oil-free compressors developed to provide high quality air for instrumentation and control, surface demand diving, breathing air for hazardous environments, in desert and arctic areas and marine applications.

This series can be supplied in one, two or four cylinder versions with 5 inch bores and stroke. Cylinders are all fitted with carbon rings. The one and two cylinder units are vertical, while the four cylinder versions are in vee form. Crankcase motionwork is force lubricated in all cases and is fitted with integral air-blast after coolers.



Series 175
Outputs up to 35/40 cfm
Pressures up to 2,500/4,000 psi

Series 175 are rugged, high output compressors suitable for both air and gas duties. Used for pneumatic gun operation for offshore seismic surveys. With associated filtration and drying equipment also widely used for breathing air bottle charging and for handling oxyhelium mixtures for deep diving.

Series 175 is a vee-form, four cylinder, three stage compressor with water or radiator cooling, incorporating inter and after coolers and moisture separation.

A filter is a device for removing oil that need never have been there.

No filter removes all the oil from the air — all of the time. No matter how often you change it.

And that's a principle that needs changing in itself.

A filter, in fact, needn't be there in the first place.

JOY™ made them unnecessary by just not allowing oil into the compressor cylinder. It stops in the crankcase because Joy's double acting cylinder head design uses an extra long piston rod and spacer. That part of the rod which runs in the crankcase never enters the cylinder. Fitting the long rod with an oil baffle ring and a stack of segmented wiper rings ensures no oil carry over or creep.

Result? A dry cylinder, which, with some specially designed corrosion resistant components gives a capacity for hard wear that makes the argument for lubrication seize up.

The savings on expensive filter elements soon make up the price difference between a lubricated and an oil-free machine.

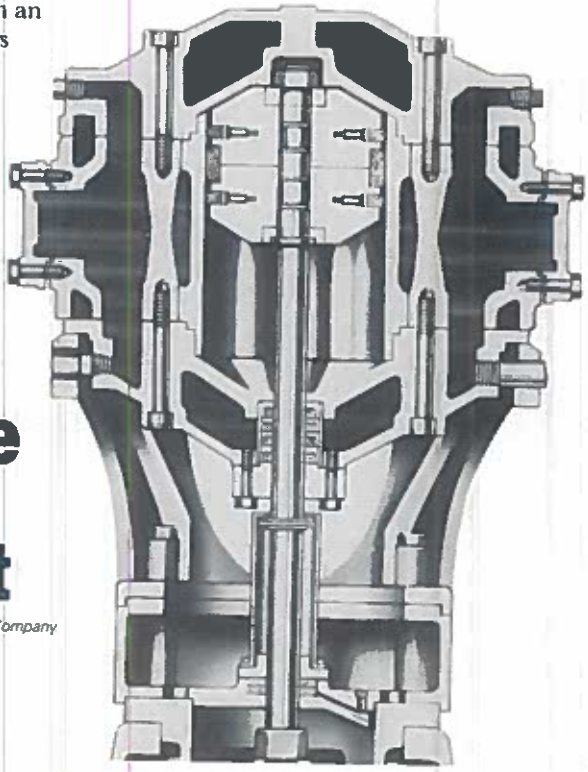
So give up changing the filter.
Change to JOY™

Where the oil isn't

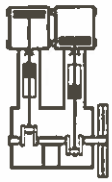
Send for our booklet on JOY™ Oil-Free Compressors. It contains all the facts.

Joy Manufacturing Company (UK) Limited
Peel Park Place, East Kilbride, Glasgow G74 5LN.
Telephone: 03552 39121

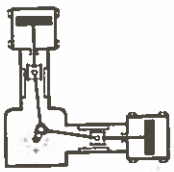
JOY™



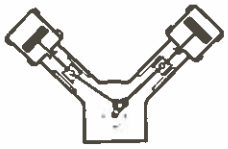
'JOY' and the Joy world logo are trade marks of Joy Manufacturing Company



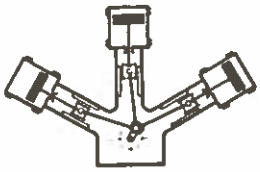
In line



L-type

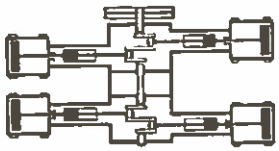


V-type

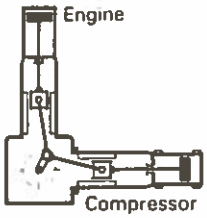


W-type

Fig 2

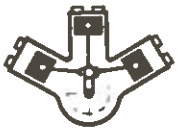


Horizontal opposed

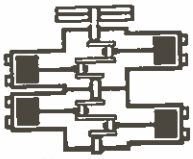


Integral L-type

V-type

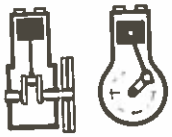


W-type

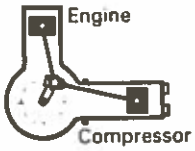


Horizontal opposed
(Boxer type)

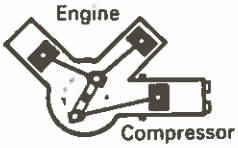
Fig 3



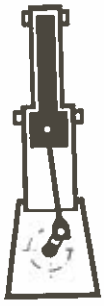
Vertical



Integral L-type



Integral W-type



Vertical with stepped piston
(Two-stage)

Horizontal with stepped piston
(Four-stage)

Fig 4

The use of multi-stage reciprocating compressors is normally confined to higher pressure requirements. Three-stage units for continuous duty are produced capable of discharge pressures up to 175 bar (2500 lb/in^2), whilst output pressure can be further increased, if necessary, by additional staging. The design tends to be much more complex than in the case of single- and two-stage reciprocating compressors, but again may embrace either single- or double-acting cylinders and trunk pistons or crosshead configuration.

For continuous duty compression ratio per stage is usually of the order of 2.5 to 3.5. Compression ratios of 6 or more per stage may, however, be used on multi-stage machines intended primarily for intermittent duties.

Labyrinth Piston Compressors

This description is given to the type of non-lubricated reciprocating compressor working without piston rings, employing instead an extended labyrinth seal. This is formed by fine grooves cut in the cylinder wall and a coarser screw thread form cut in the piston skirt. The piston rod seal is also usually of labyrinth type. Delivery is oil-free but internal leakage is higher than with ringed piston types.

Rotary Screw Compressors

Rotary screw compressors employ two intermeshing rotors with helical lobes. The inlet is located at one end of the rotors. As the rotors revolve, the space between the unmeshing lobes increases, allowing inlet air to fill up the intervening space, until the male lobe is disengaged from the female lobe along its whole length. The helices of the male and female rotors are designed to permit complete charging of the inter-lobe space before they remesh. On completion of the filling operation, the inlet ends of the male and female rotors pass the inlet port and become sealed in the casing. With continuing rotation, the male and female lobes begin to re-engage each other, the volume of this space is reduced and compression begins. This compression increases with continuing rotation until the built-in pressure ratio is reached and the air is discharged through the outlet port, see Fig 5.

The absence of valves and unbalanced mechanical forces enables screw compressors to run at high shaft speeds. In dry type screw compressors the rotors are driven by external timing gears.

Clearances between the male and female lobes, between lobes and casing, and between the ends of the rotors, are kept to the minimum, so that the minimum of slip or leak-back from the high pressure side to the inlet side is obtained. This machine has the advantage over the reciprocating

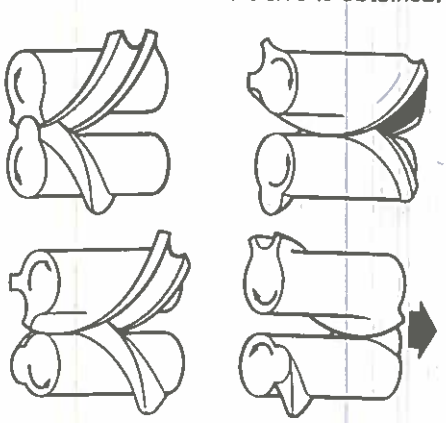


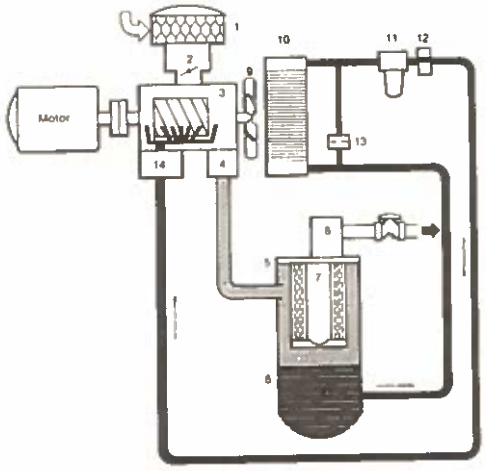
Fig 5 Working principle of a rotary screw compressor.

compressor of mechanical simplicity, as there is no contact between the male and female rotors and/or the casing. Therefore no internal lubrication is required, and oil-free air is obtained. In addition, as inlet and discharge valves are not required, maintenance is greatly simplified. A particular limitation of dry screw compressors, however, is that very high speeds are needed to maintain compressor efficiency at low capacities.

This limitation is overcome in the oil-flooded rotary screw compressor where oil is injected into the clearance space to seal the internal clearances. This oil also serves to lubricate the rotors, one of which can drive the other rotor (*ie* eliminating the timing gears); and serves a further function in providing cooling of the air during compression.

A typical oil-flooding system is shown in Fig 6. Injected oil is reclaimed in two stages, and recirculated. Injection is normally done by using the discharge air pressure, although on some designs it may be provided by a separate scavenging pump.

This type of rotary screw compressor is also used for supplying oil-free air, using water-injection instead of oil-injection. The disadvantages in this case are the necessity of protecting the compressor interior parts from rust, and the need for special seals to separate the compression space from any oil-lubricated bearings.



1 – air filter
2 – inlet throttle
3 – compressor element
4 – non-return valve
5 – oil separator
6 – oil sump
7 – oil separating filter
8 – non-return valve
9 – cooling fan
10 – oil cooler
11 – oil filter
12 – throttling
13 – thermostatic valve
14 – non-return valve

Fig 6 Oil-flooding system for a rotary screw compressor. (Atlas Copco)

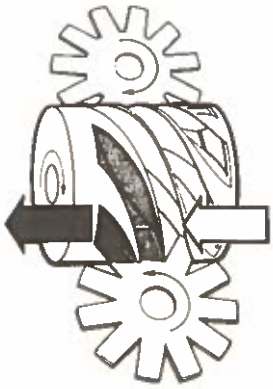


Fig 7 Working principle of a single-screw rotor compressor. (Atlas Copco).

Capacity regulation can be provided by a valve that controls the air supply to the compressor, or more simply by varying the speed. The delivery pipe is normally fitted with a check valve so that the system is automatically closed when the compressor is stopped or idling. In the latter case air is conducted from the high-pressure side to the intake side via a suitable channel to prevent the compressor working as a motor.

A further type of oil-flooded screw compressor is shown in Fig 7. This design employs a single screw with two gate rotors providing tight sealing. In this case the pressure ratio is determined by the number of screw threads.

Screw compressors are suitable for high speed operation and thus deliver high capacities from a small machine. They can be used to generate much higher pressures than a Roots blower. Pressure ratios of 3.5:1 to 5:1 can be obtained in a single stage and 12:1 in two-stage machines. Thus single-stage machines are suitable for delivery pressures up to 4.2 bar (60 lb/in^2) and two-stage machines for deliveries up to 10 bar (150 lb/in^2). Operating speeds are normally high, since this enables the size of the machine to be reduced, and may range from 3 000 to 25 000 rev/min. Typical capacity range is from 2.8 m^3/min (100 ft^3/min) FAD up to 425 m^3/min (15 000 ft^3/min) FAD.

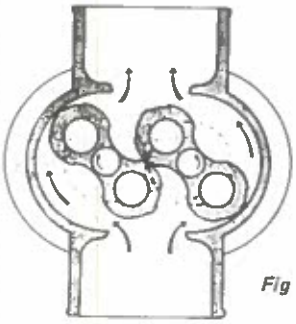


Fig 8

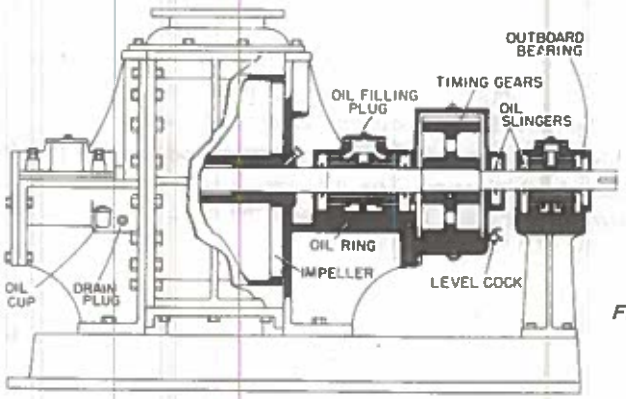


Fig 9

Lobe Rotor Compressors (Figs 8 and 9).

Lobe rotor compressors (Fig 9) operate on a similar principle to dry type rotary screw compressors, but are restricted to the generation of low pressures only.

Volumetric efficiency depends on maintaining the smallest possible clearances between the rotors and between rotors and casing. Delivery pressure depends on the back pressure in the system since no actual compression takes place in the compressor. This normally results in poor efficiencies at pressures above 0.6 to 0.7 bar (8–10 lb/in^2) although pressures up to 2 bar (30 lb/in^2) can be obtained with reasonable efficiency on two-stage machines. Special versions of Roots blowers may be capable of providing pressure of up to 3 bar (45 lb/in^2).

The principal advantages of this form of blower are simple construction and high operational reliability – particularly for the low pressure delivery of air or gases.

Speed of operation of Roots blowers ranges from about 250 rev/min for large machines to 6 000 rev/min for small i/c engine superchargers. High speeds are usually desirable in order to reduce the size of the machine and increase volumetric efficiency, leakage rate being largely independent of the actual size of the machine, but increasing substantially with increasing pressure. Back pressure or backflow is also responsible for a large rise in temperature relative to the rise in pressure.

At delivery pressures up to about 0.6 bar (8 lb/in^2) efficiencies compare favourably with most other types of compressor, but due to increasing losses the maximum pressure obtainable with a single stage is of the order of 1 bar (15 lb/in^2). Two-stage machines, used for pressures up to 2 bar (30 lb/in^2), are normally provided with intercooling to maintain a reasonable discharge temperature. Low pressure (under 1 bar) Root's blowers are available with capacities from 1.4–8.5 m^3/min (50–300 ft^3/min).

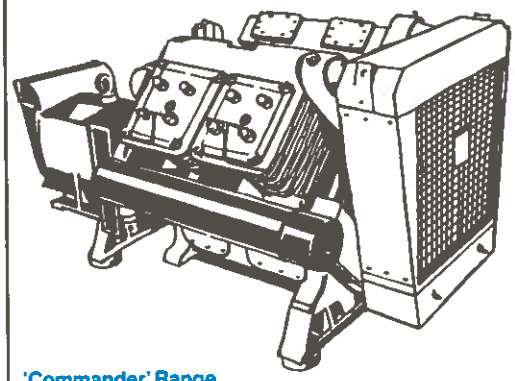
NEI -A.P.E. Ltd

Belliss & Morcom

The total service in service air

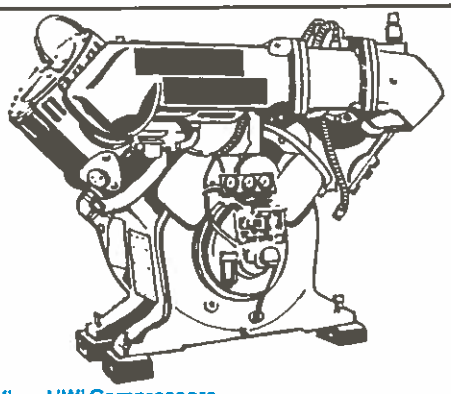
Belliss & Morcom offer a total service in medium to heavy-duty service air.

Air or water cooled compressors are available, lubricated or oil-free in capacities from 120 to 3000 cfm (3.4 to 84 M^3/min). They can be skid-mounted, free-standing or supplied as completely packaged units. The outstanding qualities of the range are massive dependability, easy installation and maintenance and a rationalised build-up which facilitates servicing with speed and economy.



'Commander' Range

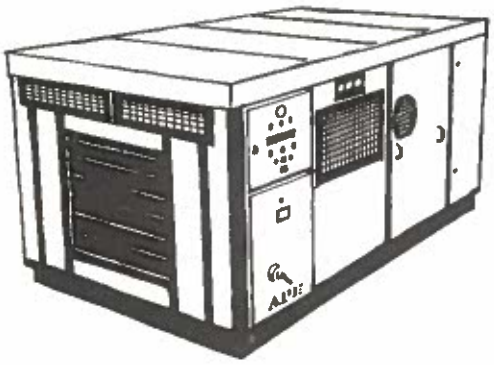
Air-cooled medium duty packaged compressor for easy instant installation. 3 models available: 2 V-twins (300 and 420 cfm) and 1 V-four cylinder models (600 cfm). All models are designed for continuous duty – inside, outside or even underground in a flameproof version.



'V' and 'W' Compressors

Conforming to metric standards these compressors incorporate every advance made possible by modern technology.

They are renowned for long-term dependability and low-vibration, eliminating the need for special foundations. Available lubricated or oil-free, in single-stage, two-stage and three-stage form.



New PBM Series Medium Duty Packaged Compressors

Models provide outputs of 300 and 420 cfm respectively under designations PBM 30/42/60. These are acoustically packaged compressors, air-cooled and self-contained for simple installation. The silencing is extremely effective making these units compatible with most working environments.

New PVH Series Heavy-Duty Packaged Compressors

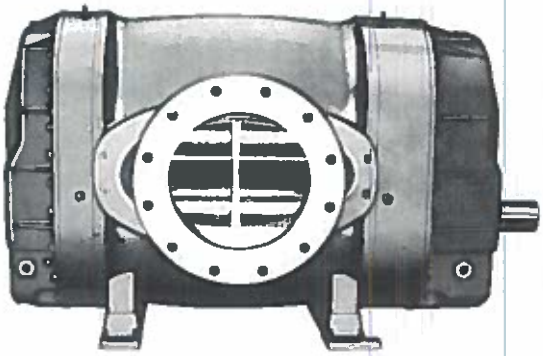
Acoustically packaged with all ancillaries these units represent a major advance for users of heavy-duty service air. No special foundations are needed. Silencing is most effective. Capacities from 500 to 1200 cfm.

Belliss & Morcom

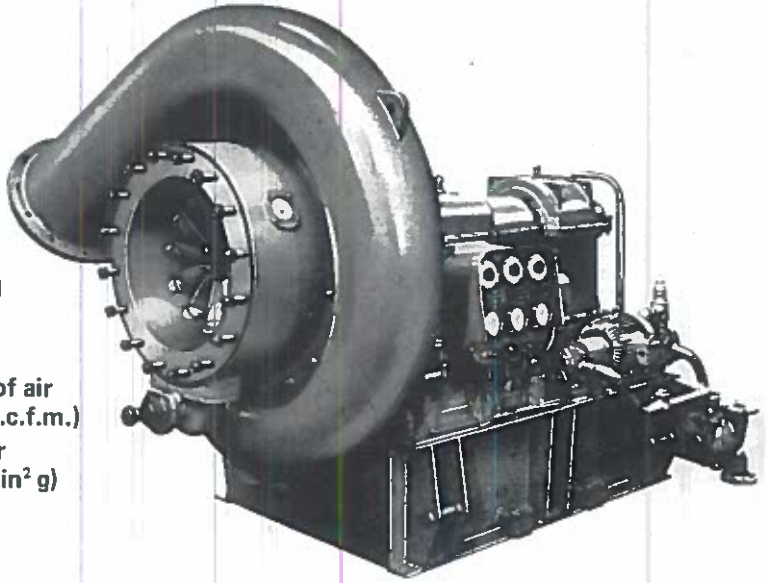
NEI -A.P.E. Ltd A Northern Engineering Industries Company

Icknield Square, Birmingham B16 0QL. Telephone: 021-454 3531. Telex: 337507. Cables & Telegrams: APEBIR G.

compressors--exhausters

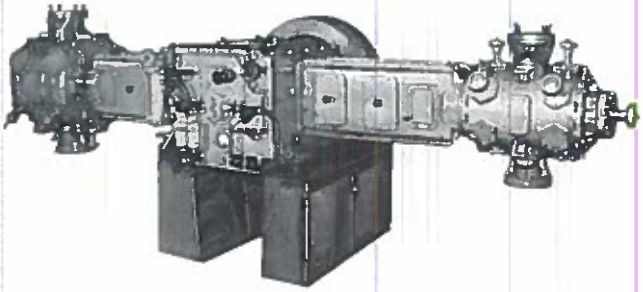


The Donkin MkII Roots type Compressor is an improved range of positive displacement machines. It is designed to give a pressure rise of 1.03 bar (15 lbf/m^2) from atmosphere on all sizes with a capacity range of 16.8 m^3/m (600 cfm) to 510 m^3/m (18,200 cfm). As an exhauster, vacuums up to 380 mm Hg (15" Hg) can be achieved under normal dry operation and 560 mm Hg (22" Hg) with water injection to the inlet side.



An SD90C Centrifugal Blower as supplied for a pulp and paper mill in Canada. It is driven by an electric motor through epicyclic speed increasing gear.
Typical duty:

Volume flow 820 m^3/m of air (29,000 s.c.f.m.)
Working pressure 0.55 bar (8.0 lbf/in^2 g)
Blower speed 7090 r.p.m.



A 2HB2GT two stage, oil free, dual duty argon/nitrogen compressor. Typical duty:-
Pe 0.02 Kgf/cm^2 (0.3 lbf/in^2)
Pa 3.18 Kgf/cm^2 (45.3 lbf/in^2)
Capacity 4.4x1162 Nm^3/hr (10^4 s.c.f.h.)

BRYAN DONKIN COMPANY Ltd; CHESTERFIELD. Phone (0246) 73153 & 70151

valves and regulators··

Fig.994 Hot Gas Valve.
This valve is one of the very few valaves available for continuous duty at elevated temperatures, especially in the size ranges offered.
2" to 42" bore diameter.
Working pressure 20 lbf/in².
Temperature 750°C.

Donkin 680 Regulator.
An extremely versatile, sensitive and stable regulator giving 40% extra capacity.
Sizes 2" to 8" bore.
Maximum pressure —
Inlet 30 lbf/in²
Outlet 4" to 56" w.g.

Non-Return Valve.
Size range 2" to 48" diam.
Working pressure 100 lbf/in²
(7.03 Kgf/cm²)

Fig. 50 Ball Valve.
A quick operating valve with wide field of application where 100% cut-off is required.
Especially suitable for the isolation of gases.
Sizes ½" to 6".

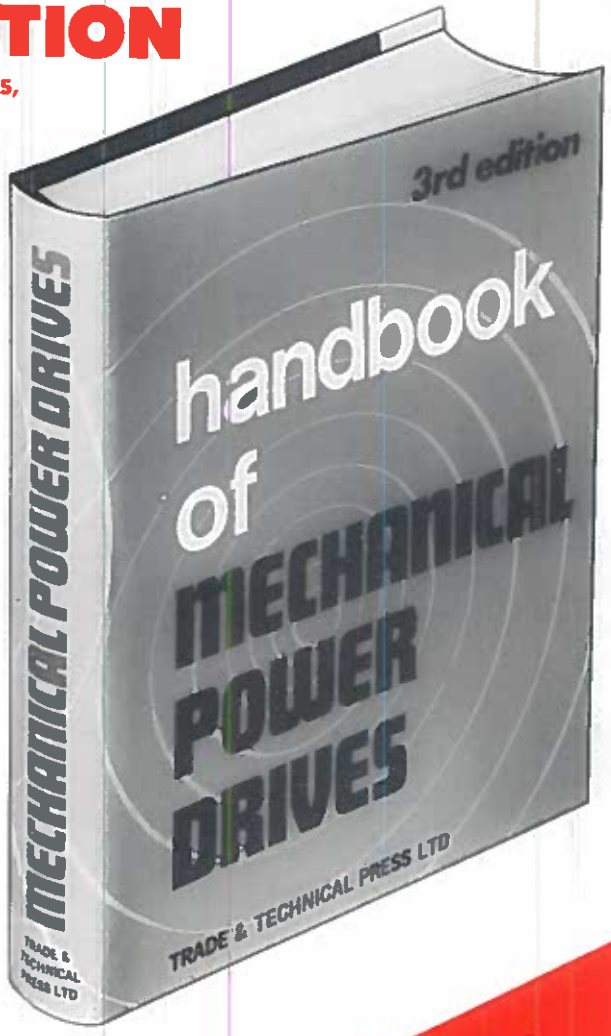
donkin

valves, regulators and compression plant for gas and process installations

BRYAN DONKIN COMPANY Ltd; CHESTERFIELD. Phone(0246) 73153 & 70151

NEW EDITION

680 PAGES. 1500 DIAGRAMS, ILLUSTRATIONS, TABLES, CHARTS, ETC., INCLUDING BRITISH STANDARD AND METRIC EQUIVALENTS.



SECTION 1 — Systems and Fundamentals: The Energy Crisis; Prime Movers and Drivers; Electric Motors; Electric Motor Speed Controls; Servo Motors and Controls; Mechanical Power Transmission; Bearings (Principles); Shafting (Principles); Belt Drives (Principles); Gear Drives (Principles); Variable Speed Drives, Vibration, Shock and Noise; Machinery Health Monitoring; SI Units; Power Transmission Selection Guides.

SECTION 2 — Power Transmission Equipment: Bearing Loads; Plain Bearings; Ball, Roller and Needle Bearings; Plummer Blocks; Elastomeric Bearings; Miscellaneous and Special Bearings; Flat Belts; V-Belts; Special Belts; Installation and Maintenance of Belt Drives; Chain Drives, Belt and Chain Tensioners; Couplings, Clutches; Brakes; Clutch and Brake Calculations; Freewheels; Gears and Gearing; Mechanical Transmissions; Magnetic Drives; Fluid Drives.

SECTION 3A — Related/Associated Mechanisms and Elements: Universal Joints; Flexible Shafts; Pulleys (Sheaves); Dampers and Shock Absorbers; Torque Limiters; Tension Stabilizers; Keys and Splines. Circlips and Retaining Rings; Ball Screws and Lead Screws; Springs; Miscellaneous Mechanisms.

SECTION 3B — Related/Associated Systems: Speed Reducers; Step Motors; Dynamic Braking; Conveyor Belt Drives; Rope Drives.

SECTION 3C — Ancillaries and Related Subjects: Lubrication and Lubricants; Filters; Seals; Oil Seals; Instruments; Anti-Vibration Treatment; Noise Control.

SECTION 4: British Standards Relevant to the Power Transmission Field.

SECTION 5: Buyers' Guide; Editorial Index.

TRADE & TECHNICAL PRESS LTD.,
CROWN HOUSE, MORDEN, SURREY SM4 5EW, ENGLAND.

ORDER YOUR COPY NOW

Vane Compressors

There are numerous variations on the design of vane compressors, but all follow the general form of a cylindrical rotor with a series of radial slots cut into it. The rotor is mounted eccentrically in a cylinder. Each slot in the rotor carries a blade, the outer edge of which is kept in contact with the cylinder by centrifugal force. As the rotor turns, the space between each pair of adjacent vanes alternatively grows and contracts. The inlet port is positioned to let in the air where the space is opening up, and the delivery port opens where the space closes. This cycle of operation is shown in Fig 10.

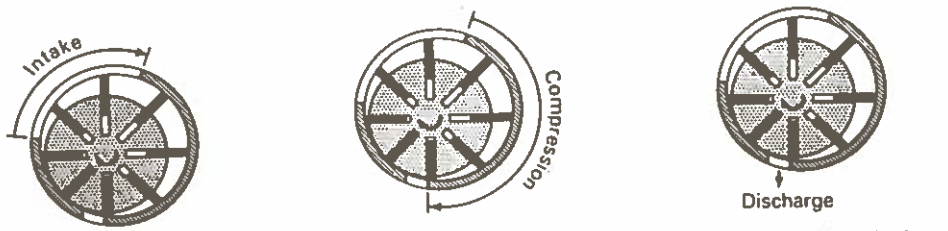

Intake
Compression
Discharge
As rotor turns, gas is trapped in pockets formed by vanes
Gas is gradually compressed as pockets get smaller
Compressed gas is pushed out through discharge port

Fig 10 Working principle of a rotary vane compressor.

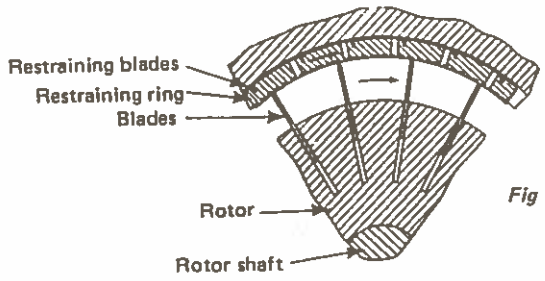


Fig 11 Vane compressor with sliding rings

Vane material is commonly resin-impregnated fabric laminate, but on smaller machines may be of hardened steel. For oil-free delivery, bronze and carbon/graphite vanes are normally used.

Due to the large centrifugal force acting on the cylinder, the larger machines usually have a floating cylinder liner or rings, which are slightly less in diameter than the cylinder bore. Consequently the blades are in contact with this floating member, which is free to rotate in the cylinder, with the result that in theory all wear between the blades and cylinder is eliminated.

Medium or large capacity vane compressors running at high speeds normally incorporate a method of avoiding contact between the vanes and casing wall in order to reduce wear and minimize frictional losses and overheating. A widely used method is to fit restraining or floating rings over the vanes, the internal diameter of these rings being slightly less than the cylinder bore. The geometric difference is calculated to preserve a minimum clearance between the vane tips and the casing walls under operating conditions — see Fig 11. The rings actually rotate with the vanes but the peripheral speed of each vane tip varies with the degree of extension. The rings rotate at constant speed. There is thus some relative motion between vane and rings, but the contact between rotor and rings is essentially a rolling motion.

TABLE I – SUMMARY OF COMPRESSOR CHARACTERISTICS

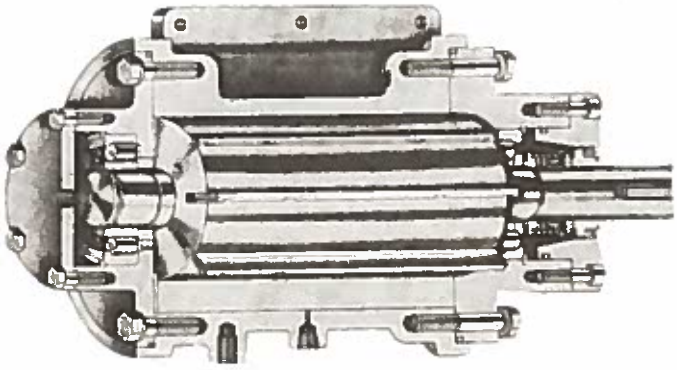
Type		Maximum Pressure (gauge) bar	Maximum Pressure (gauge) lb/in^2	rev/min	Compression Ratio	Maximum Capacity or Range m^3/min	Maximum Capacity or Range ft^3/min	Remarks
RECIPROCATING	Single-stage	10	150	up to 1 500	2.5 to 3.5 per stage			Single or double acting
	2-stage	70	1 000					Multi-cylinder in various configurations
	3-stage	175	2 500					
	4-stage	350	5 000					
	5-stage	500	7000					
	6-stage	550 to 700	8 000 to 10 000		Up to 6 per stage for intermittent operation			
ROTARY	Screw, single-stage	up to 4	up to 60	3 000 to 25 000	5:1 (single-stage) 12:1 (2-stage)	3–140	100–5 000	Compression ratio 3.5:1 to 5:1
	Screw, 2-stage	up to 10	up to 150			3–420	100–15 000	Compression ratio up to 12:1
	Screw, oil flooded	10	150	3 000 upwards		2.8–425	100–15 000	Single- and two-stage
	Roots, single-stage	up to 1	up to 15	250 (large)		8.5	300	Used mainly as blowers
	Roots, 2-stage	up to 2	up to 30	up to 6 000 (small)				
VANE	Single-stage	4	60	250 (large)		1.4–5.7	50–200	Water cooled above 30 lb/in^2
	2-stage	up to 10	up to 150	up to 3 500 (small)		1 400	50 000	
	Oil-cooled	up to 8.5	up to 120	up to 3 000				Higher efficiency than 'dry' types
LIQUID RING	Single	up to 1.5	up to 20	up to 3 600		0.5–80	17–2800	Low volumetric efficiency
	Series (2-stage)	up to 2–3	up to 30–45					
DIAPHRAGM	Mechanical, single-stage	4	60			0.7	25	
	2-stage	7	100			0.2	6	
	Hydraulic, single-stage	19	220			1.7	60	
	2-stage	1 400	20 000			0.6	20	
	3-stage	2 100	30 000			0.4	15	

The number of blades is usually limited to a maximum of 10 or 12 with high speed designs providing discharge pressures of the order of 7 bar (100 lb/in^2). A lesser number – *eg* 8 blades – may be necessary where relatively thick blades are used. To increase volumetric efficiency it is also desirable to spread the pressure difference between inlet and outlet over two rotors, leading to a two-stage rotor. It should be noted that, unlike other compressor types, intercooling is not required between stages (since continuous cooling is provided by the oil in both stages) and the only significance of staging is in improving volumetric efficiency.

Vane compressors do not have valves, but a check valve may be installed after the compressor. Capacity may be controlled by a throttle valve in the air supply line, although automatic stop–start regulation is usually more economical.

Vane-type compressors may be single- or two-stage, cooled by water, air or oil. Single-stage vane pumps are generally used for pressures up to 4 bar (60 lb/in^2) and two-stage machines for pressures up to 10 bar (150 lb/in^2) whilst operating speeds may range from as low as 250 rev/min up to 3500 rev/min, speed normally being inversely proportional to size. Being a well balanced machine, a vane compressor is well suited for direct connection to a relatively high speed driver (*eg* an electric motor). Its efficiency, however, is generally less than that of a reciprocating compressor and can decrease substantially with age through blade wear.

The vane compressor is generally most efficient at lower working pressures. Capacity range covered by vane compressors is normally 1.5–6 m^3/min (50–200 ft^3/min).



Single-stage vane compressor.

Cooling Vane Compressors

On earlier designs cooling was either by air or water. In the former the casing is usually finned to increase the external surface area and thus the rate of heat dissipation. With water cooling, the water jacket is normally cast integral with the casing and a circulatory flow of clean cold water established so that the temperature of the discharged air does not exceed 150°C (300°F). Cooling requirements are not as severe as with reciprocating compressors since there are no restrictions on the discharge side and incoming air does not come into contact with a heated surface to raise its initial temperature. Water cooling is normal for vane-type compressors where the discharge pressure exceeds 2 bar (30 lb/in^2). In the case of two-stage machines, intercooling is normally provided, either by a separate intercooler, as in the case of reciprocating compressors, or by an intercooler built into the base of the machine.

Oil-cooled rotary vane-type compressors are a more recent development, the principle being that oil is injected in relatively large quantities into the compressor and recovered on the discharge side and recirculated. Besides providing the most effective form of cooling by acting as a heat transfer medium in direct contact with the air, flooding the machine with oil considerably reduces internal leakage and improves volumetric efficiency. Volumetric efficiencies of oil-cooled vane compressors, in fact, are usually directly comparable with those of reciprocating machines, whilst the former are considerably more compact and lighter for a similar duty.

The quantity of oil circulated and injected is normally designed to provide the maximum acceptable degree of cooling and in some cases may be of the same order as the volume of air delivered at the discharge end. To remove the oil at the discharge end it is usual first to interrupt the flow with baffles on which the oil can impinge, then to pass the discharge through a separator or filter to remove as much as possible of the remaining oil. Oil separation is assisted by the fact that the discharge temperature is low so the oil is in the form of droplets rather than aerosol size particles and vapour, and oil removal to the order of 99.0–99.9% is usually possible by simple mechanical separation.

Diaphragm Compressors

Diaphragm compressors are positive-displacement, oil-free type. They are generally classified as either mechanically driven or hydraulically actuated, depending on the method of deriving diaphragm movement. Mechanical diaphragm compressors are manufactured in smaller capacities than hydraulically actuated units and consequently overlapping of the two types only occurs over a restricted range.

Mechanical machines are less expensive to produce, are very compact and can be used on applications involving sub-atmospheric pressures. (See also Fig 12).

Hydraulically actuated units are able more easily to generate high pressure than are mechanical machines as the latter are usually limited by considerations such as bearing loading.

Apart from the fact that the compact form of a diaphragm unit enables it to be built directly onto an electric motor or driver (or hydraulic pump in the case of a hydraulically-operated machine), the main advantages offered by diaphragm compressors are:

(i) Complete gas-tightness with only static sealing involved.

(ii) Isolation of the gas being handled and thus the possibility of 100% oil-free compressed gas supply.

The main limitations of the type are low delivery rates and a limited compression ratio. The latter applies particularly to mechanically-operated diaphragms where the diaphragm is normally of synthetic rubber. Hydraulically-operated diaphragms may employ metallic diaphragms and be capable of generating much higher pressures. Multi-staging is possible with both types.

A further method of obtaining high pressure working with a mechanically-operated diaphragm compressor is to enclose the compressor and driver within a pressure vessel. This vessel is pressurized to a level equal to that present at the inlet. The same pressure is thus present on the underside of the diaphragm, reducing the pressure differential on the diaphragm when working.

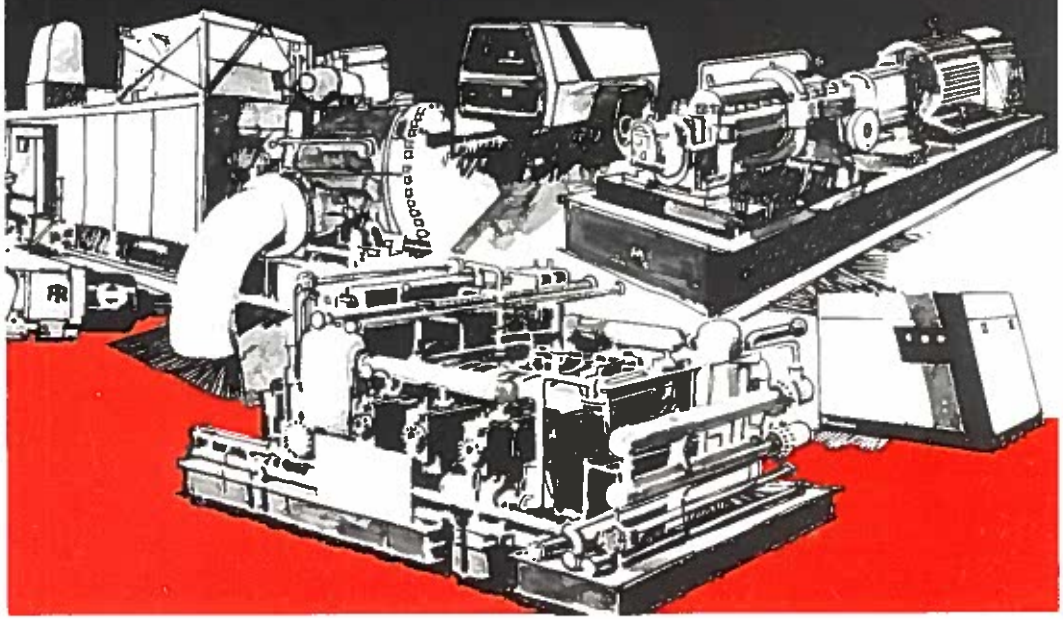
Liquid Ring Compressors

The liquid-ring compressor is another positive-displacement oil-free type. It is based on a rotor with a series of fixed blades, mounted in a cylinder and so arranged that clearance between the blade tip and the cylinder varies cyclically on each revolution of the rotor – Fig 13. The cylinder is part-filled with a liquid, usually water when the machine is handling air. During operation, the

Equipment and
technical developments
for applications throughout
the world's
energy industries

INGERSOLL-RAND

Bowater House, Knightsbridge, London, SW1X 7LU
Telephone: 01-584 5070 Telex: 915813-4



"NORTHEY"
TRADE MARK

Oil- and
Particle-free
AIR and GAS

COMPRESSORS

"NORTHEY"
TRADE MARK

BOOSTERS

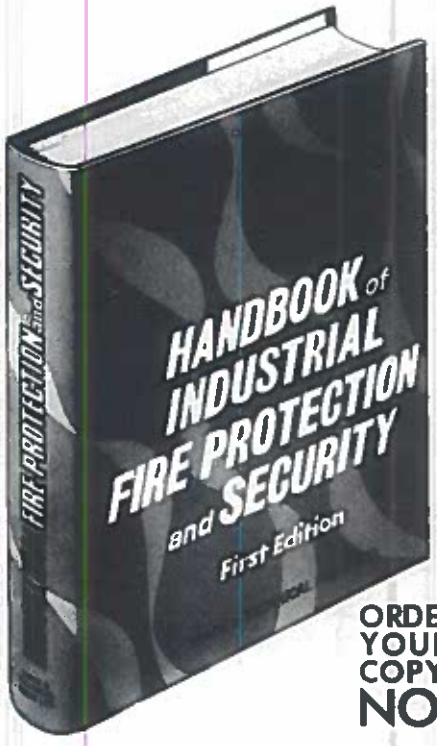
"NORTHEY"
TRADE MARK

VACUUM PUMPS

Northey
ROTARY COMPRESSORS LTD.

ALDER ROAD, PARKSTONE,
POOLE, DORSET, BH12 4BB

Tel : PARKSTONE (0202) 741725 Telex: 41444

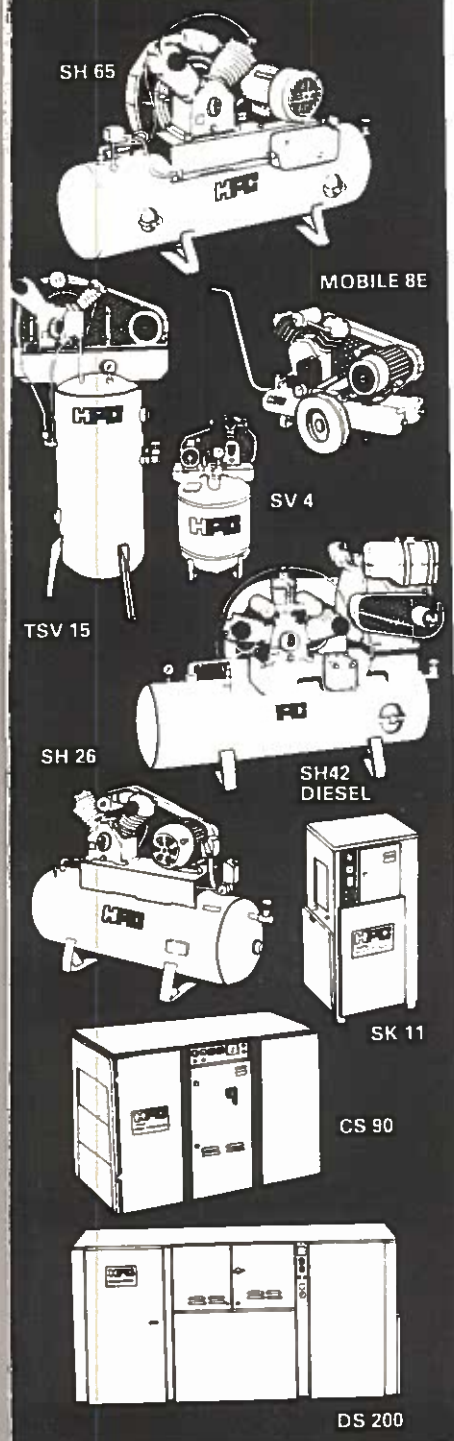


ORDER
YOUR
COPY
NOW!

A 600 page Handbook, specially produced for those concerned with the safety and security of factories, works, warehouses and offices. A complete reference work, of paramount importance in these days of high fire risk, breaking and entering, vandalism, arson and political anarchy. Contents include:- 1. Fire: Fire Prevention; Fire Training; Fire Engineering; Fire Alarms; Fire Fighting Equipment Appliances; Fire Extinguishers — all types; CO_2 Installations; Fire Protection Equipment, Blankets, Safety Clothing, Eye Protection, etc, etc. 2. Hardware: Locks; Grilles; Safes, Strongrooms; Safety Containers, etc. 3. Detection: Fire Alarms; Fire Detection Systems; Heat detectors; Smoke Detectors; Gas and Fume Detectors; Burglar Alarms. 4. Security: Industrial Security; Alarm Systems; Guard Systems; Private Police Forces; Security Vehicles. 5. Buildings: Materials; Fire Resistant Structures; Fire Ventilation; Fire Doors; Escape Systems. 6. Materials: Fire Resistance; Fire-resistant and Non-Combustible Materials; Fireproof Materials; Fireproofing of Materials; Plastic in Fires. 7. Environmental: Electrical Fires; Hazardous Environments; Hazardous liquids; Hazardous Stores. 8. Emergency: First Aid; Ambulances; Escape Equipment; Rescue Equipment; Emergency Lighting and Power; Emergency Services; Communications Equipment; Public Address Equipment. 9. Reference: Industrial Fire Brigades; Fire Prevention Acts and Regulations; National Authorities; Publicity Material; Rescue Corps; Salvage and Wrecker Services; Security Services. 10. Trade Names Index, Classified list of equipment and materials, manufacturers, etc.

TRADE & TECHNICAL PRESS LTD.
CROWN HOUSE, MORDEN, SURREY

A comprehensive range of air compressors



HPC

PISTON COMPRESSORS

A range of quality horizontal receiver-mounted and base mounted compressors from ½hp to 50hp. Pressures up to 200psi (13.8 bar).

For over 21 years HPC have been producing air compressors to the highest technical standards in the belief that reliability and long life are the prime qualities required. Steady development over this long period has resulted in a range of compressors tailored to meet the specific requirements of almost every industrial user.

Of equal importance is the competent and enthusiastic HPC team who ensure that all enquiries are dealt with efficiently and that the correct compressor arrives when and where required.

Send for colour brochure giving full details of range.

PLUSAIR ROTARY SCREW COMPRESSORS

Quiet, efficient and low on maintenance costs, PLUSAIR screw compressors from 10hp to 335hp. Pressures up to 188psi (13 bar).

INSTALLATION & AFTER SALES SERVICE THROUGHOUT THE U.K.

HPC AIR COMPRESSORS

HPC Engineering Limited, Victoria Gardens, Industrial Estate, Burgess Hill, West Sussex Telephone (04446) 41671 Telex 87606

One Order **One Account**

why go to supplier?

A for valves and cylinders
B for air service equipment
C for air tools
D for tube and fittings
E for air compressors

when you can contact

THORITE

Yorkshire's compressed air specialists

For all your requirements ***Simple as ABC***
Phone: Bradford 663471/6 Leeds 444554 or Huddersfield 34245/6 or Doncaster 25492/3

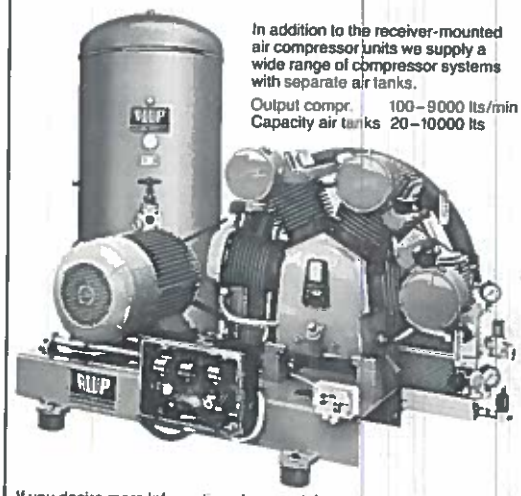
Thomas Wright (Bradford) Ltd.
Thorite House
Laisterdyke, Bradford BD4 8BZ
Phone 663471/6 Telex 51674
code, Thorite Bradford

Thomas Wright (Huddersfield) Ltd.
Long Lane, Moldgreen
Huddersfield HD5 9LY
Phone 34245/6

Thorite Pneumatics Ltd.
Lowfields Road
Leeds LS12 6SB
Phone 444554
Telex 557628

Thorite Pneumatics Ltd
Churchill Road
Doncaster DN1 2TF
Phone 0302 25492/3
Telex 557628

Compressed Air Stations with separately installed Air Tanks



In addition to the receiver-mounted air compressor units we supply a wide range of compressor systems with separate air tanks.

Output compr. 100–9000 lts/min
Capacity air tanks 20–10000 lts

If you desire more information please ask for our detailed and free-of-charge literature.

ALUP-Kompressoren GmbH
Postfach 281 · D-7316 Köngen
Fed. Rep. Germany
Tel. (07024) 8020 · Telex 7267215

ALUP KOMPRESSOREN

Fully enclosed. Compressors with pressurized tank.

Compact. Ready for connection and operation...

... because the standard features include fully wired electronic control, motor protection and 3 metre cable with mains plug. Advantages for the user in all features.
And don't forget the favourable price-performance ratio.

Ask for detailed information. It's worth it and it doesn't cost anything. We respond straight away.

ALUP-Kompressoren GmbH
Postfach 281 · D-7316 Köngen
Fed. Rep. Germany
Tel. (07024) 8020 · Telex 7267215

ALUP KOMPRESSOREN

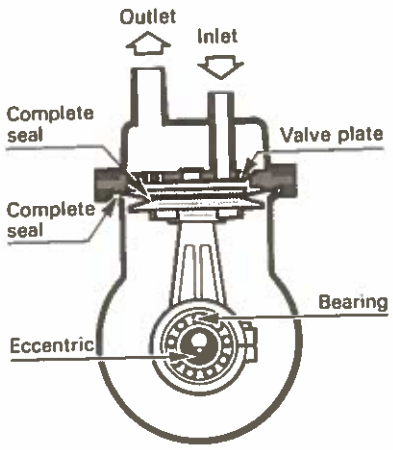


Fig 12 Cross section of a mechanically actuated diaphragm compressor.

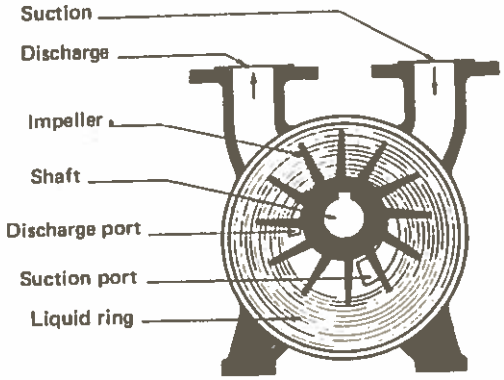


Fig 13 Cross section of a liquid ring compressor.

water is carried around the cylinder by the action of the rotor blades. Due to the centrifugal force, the water forms a solid ring around the cylinder, the inner wall of which varies in its distance from the rotor, as in fact the cylinder wall does. Thus, the space between the blades varies cyclically, as with the sliding vane compressor.

Unlike the sliding vane compressor it is possible, by suitably designing the cylinder wall, to obtain several cycles of expansion and compression per revolution. In practice this is usually limited to one or two. During operation, water may be either supplied from an outside source, or recirculated through a heat exchanger.

In theory, compression is isothermal, although in practice there is a substantial energy loss due to the churning action of the blades. The type is thus relatively inefficient (*eg* compared with a reciprocating compressor for the same duty), but due to its oil-free operation, the liquid ring compressor has applications in chemical engineering or where the air or gas being handled has corrosive elements in it. It is also used as a vacuum pump.

Liquid ring compressors are made in sizes from 1–80 m^3/min (12–940 ft^3/min). Operating speeds range from 300–3 000 rev/min depending on the size of the machine.

Dynamic Compressors

Dynamic compressors are essentially turbo-machines in which compression is achieved partially by compressing the air by a reduction in relative volume through the rotor and partly by increasing the kinetic energy of the air in respect of its absolute velocity. This kinetic energy is subsequently converted into pressure energy by diffusion in the stator. Types involved include both radial flow (centrifugal) and axial flow machines. They are, in fact, dynamic machines rather than positive-displacement devices and thus differ in characteristics from both reciprocating compressors and other types of rotary machine, working on positive displacement principles. They also differ from the other types in that no attempt is made to seal the rotor or compressing element – see Fig 14.

In the centrifugal compressor a multi-bladed impeller is rotated at high speed in a casing. Air trapped between the blades is accelerated and thrown outwards, leaving the blade tips with increased pressure and high velocity to enter a diffuser ring. Here there is a region of increasing

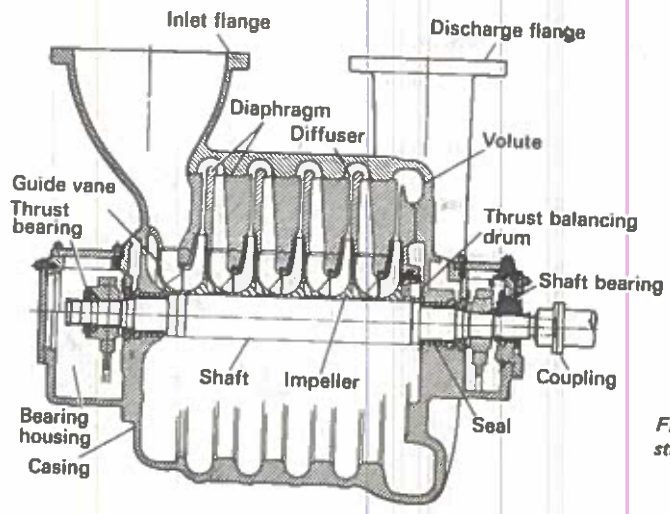


Fig 14 Sectional view of a typical five-stage horizontally split centrifugal compressor. (Atlas Copco).

area in the direction of flow, resulting in a reduction in velocity and a substantial increase in pressure. From the diffuser the air enters the volute casing with further reduction in velocity and increase in pressure. At the same time flow through the impeller is maintained by a reduction in pressure at the inlet, drawing in a continual supply of air as long as the impeller is rotating. Inlet air may enter from one side of the impeller only, or from both sides simultaneously, when the compressor is known as a single-flow or double-flow type respectively. The advantage of double-flow is that thrust forces on the impeller are balanced.

The pressure which can be obtained in a single stage is limited by the maximum permissible peripheral speed of the impeller, which is typically of the order of 300 metre/sec (1000 ft/sec). This largely limits single-stage capabilities to a maximum delivery pressure of about 0.4 bar (6 lb/in^2), although higher pressures can readily be achieved by adding more stages. Thus, typically, 2.5 bar (35 lb/in^2) may be produced in five stages and very much higher pressures can be achieved if required with further staging. Equally, mult-staging is not always associated with higher delivery pressures as multi-stage machines may be specifically designed for low pressure applications.

A distinction is also drawn between 'low' and 'high' pressure aerodynamic machines. Where the delivery pressure does not exceed about 2 bar (30 lb/in^2) such machines are generally termed 'blowers', and where delivery pressure is 2 bar (30 lb/in^2) or more they are termed 'compressors.'

Discharge pressure and volume of air delivered are co-related. Thus if at a given speed the back pressure is reduced, the volume delivered will increase, and vice-versa — see Fig 15. If the back pressure increases to a value above that generated by the compressor then a condition known as 'stall' or 'surge' occurs and gas flows back into the machine. If the pressure then drops the machine starts to deliver again and if this sequence is allowed to continue serious damage to the machine could result. Practical operating limits as regards characteristic curves, are, therefore, terminated in a surging limit — see Fig 16. To avoid surging it is desirable that the controls of the machine should incorporate a provision for isolating it from the mains when operating conditions approach the surge point, and for restoring it to operation when the surge pressure has fallen.

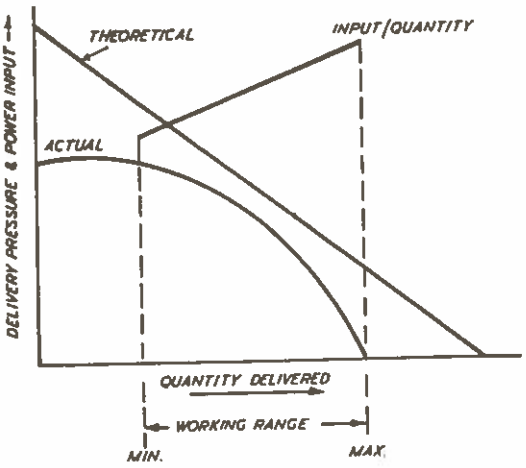


Fig 15

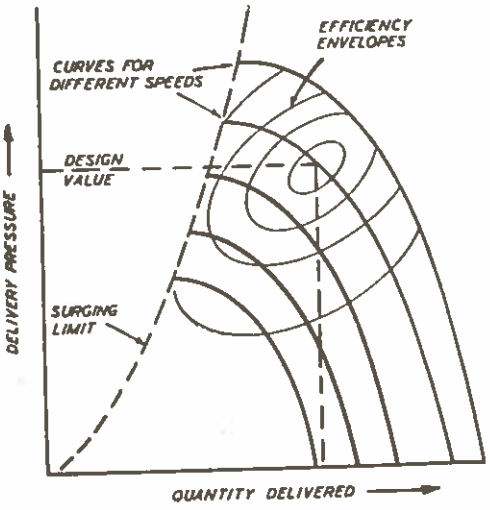


Fig 16

Operating speeds of centrifugal compressors need to be high for good performance – *eg* 20 000 rev/min upwards. Minimum capacity is mainly limited by the flow through the last stage, 160 lit/sec being a typical figure. Centrifugal compressors do not generally need cooling (except if used for generating higher pressures), but their efficiency at a normal working pressure of 6–7 bar (90–100 lb/in^2) is somewhat lower than displacement type compressors except at high capacities.

Axial Flow Compressors

Axial flow machines are similar in principle to centrifugal types except that the air is moved through a series of alternate rows of rotating and fixed blades with the mean direction of flow parallel to the axis of the rotor. In general they are more efficient than centrifugal machines, largely because of the simpler and shorter flow path, but they are considerably more susceptible to changes in throughput. Thus, whilst a centrifugal machine may tolerate reductions in throughflow of as much as 30%, axial machines can barely tolerate any reduction without serious loss of efficiency. They are particularly suited for handling large volumes of air, but centrifugal machines are better able to handle wide ranges in quantity where the variation in delivery pressure or compression ratio is relatively small.

Basic elements involved in an axial flow compressor are a row of moving blades attached to a rotor followed by a row of fixed blades attached to the casing, the pitch of the two rows of blades being in opposition – Fig 17. Curvature applied to the blades has the effect of increasing the sectional area through which the gas is flowing with resulting pressure increase as in the diffuser of a centrifugal compressor. At the same time work is being done on the gas by the rotating blades, increasing its absolute velocity. The process is repeated throughout the length of a succession of blade sets with velocity of the gas relative to the moving blades decreasing and the absolute velocity of the gas increasing. The fixed blades also act as diffusers providing a further increase in pressure to the gas passing through them. The quantity of air delivered is more influenced by delivery pressure than is the case in centrifugal machines – see Fig 18 – and the characteristic curves are somewhat flatter – see Fig 19.

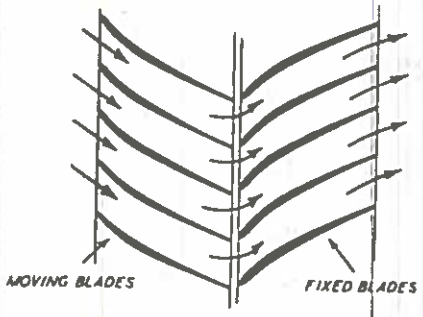


Fig 17

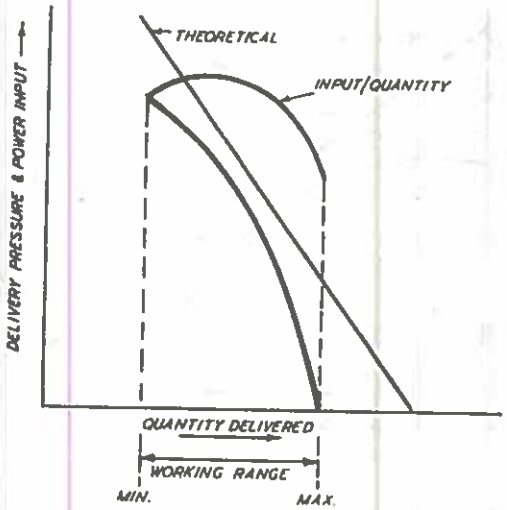


Fig 18

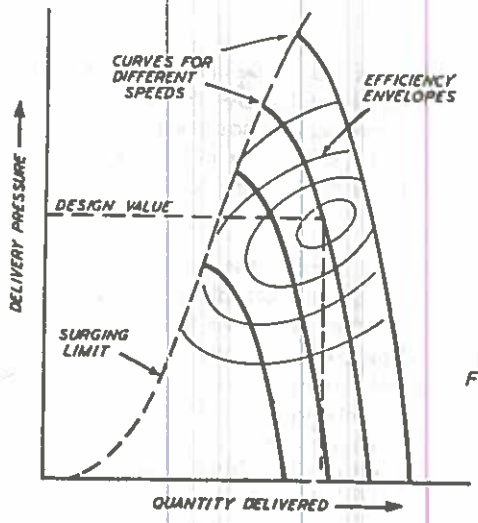


Fig 19

Axial flow machines offer a considerable economy in overall size over centrifugal machines for handling large volumes of air at high efficiency. They would be a normal choice for capacities above 65 m^3/sec and effective pressure up to about 14 bar (200 lb/in^2).

See also *Survey of Compressors.*

Compressor Performance

PRACTICAL COMPRESSORS normally work by a polytropic process, *ie* intermediate between true isothermal and true adiabatic compression – Fig 1. There are definite advantages to be gained from achieving the final degree of compression required in two or more stages, with intercooling between stages. Multi stage compression increases the volumetric efficiency and power requirements – power input being a minimum if the total work is divided equally over the stages. Cooling is particularly useful in decreasing the final air temperature and it also reduces the work input required to achieve the same degree of compression. This is exemplified in the generalized compression diagram of Fig 2 for two-stage compression and intercooling. The first stage polytropic curve approaches closely the adiabatic curve but at the conclusion of this stage air temperature is reduced to its original value by intercooling and the volume is also reduced. The second stage polytropic curve then follows more closely the isothermal line, the shaded area representing the input power saved by cooling. In practice the temperature reduction is more significant than the work-saving and the desirability of multi stage operation is greater with increasing final pressure and capacity required.

Power requirements for multi stage compression are minimal if the total work is divided evenly over the stages, although this can also be affected in practice by the degree of cooling achieved. With perfect intercooling minimum power is required when the pressure ratio of all the stages is the same.

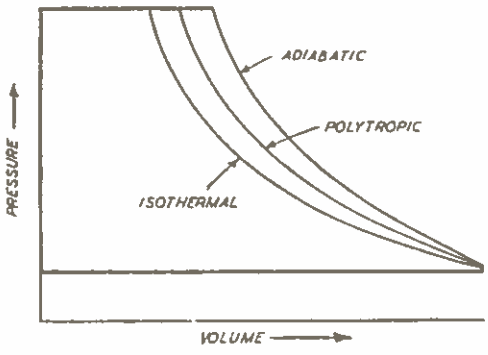


Fig 1

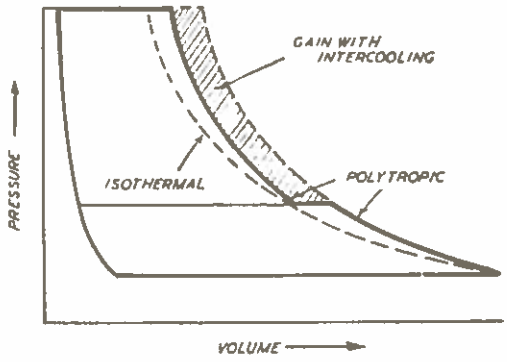


Fig 2

As a general rule, the greater the number of stages the greater the efficiency possible because of the close approach that can be realized to isothermal compression. In practice, however, the number of stages employed may have to be limited on grounds of complication and cost. Basically, therefore, for any particular pressure level there is an optimum number of stages for a compressor designed for a particular application.

Volumetric Efficiency

Volumetic efficiency is a function of capacity normally specified in terms of free air delivered (FAD). This refers to the volume of free air at the intake pressure and temperature which the compressor will take in, compress and deliver at a stated delivery pressure. Output of a compressor refers to the free air delivered when the machine is working at the specified intake pressure and temperature and specified delivery pressure.

The volumetric efficiency (η_V) is expressed by

$$\eta_V = K\left[1 - \left(V_C\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right)\right]$$

where K is a factor depending on heating of the air and valve losses (also varying with pressure ratio). A typical value is 0.96

V_C = relative clearance volume (typical values for average compressor designs 0.06–0.12)

P_1 = intake pressure of the final stage.

P_2 = discharge pressure of the first stage.

n = polytropic exponent (typical value 1.3)

Compressor Efficiency

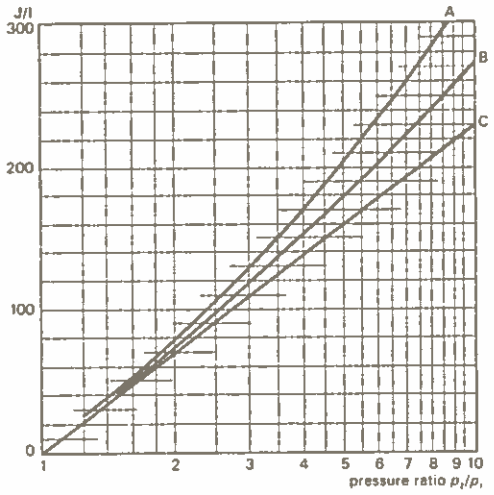
Compressor efficiency is defined by the ratio of the theoretical work required to compress a given amount of air to the actual amount of work developed on the air. The theoretical calculation may be based on isothermal compression (n = 1 in the basic formula) or adiabatic compression (n = 1.405), in which case the ratio would be referred to as the isothermal compression efficiency or adiabatic compression efficiency respectively.

The mechanical efficiency is the ratio of work done on the air to the actual work input to the machine and thus is a measure of mechanical losses. The overall efficiency is the ratio of the total power output to total power input, which can also be expressed as the product of the compression efficiency, mechanical efficiency and driver efficiency.

Thus volumetric efficiency largely governs the size of compressor required for a given duty, and overall efficiency the cost of the compressed air. Since annual power costs may approach one quarter to one third the initial cost of the compressor of a typical large volume industrial system, overall efficiency can be an important factor in deciding on the most economical plant to install.

Power Requirement

The specific energy requirement to compress unit volume of air is a direct measure of the power requirement. For stationary compressors this is in Joules per litre (J/lit); or more realistically kilowatt hours per cubic metre (kWh/m^3). This depends specifically on the pressure ratio achieved and the representative isentrop — see Fig 3.



A – single stage isentropic compression
B – two stage isentropic compression
C – isothermal compression

Fig 3 Theoretical specific energy requirement curves for air.

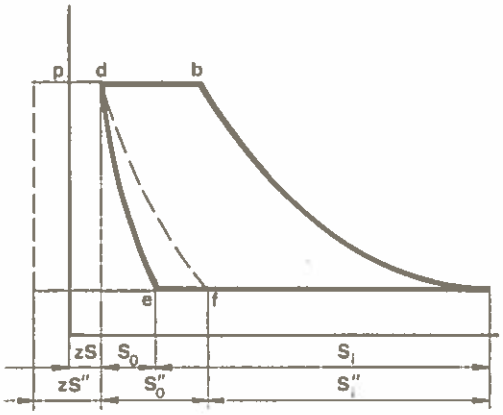


Fig 4 Ideal and practical compression diagram
Theoretical diagram abdea
Practical diagram abdfa
Suction period : theoretical S_i, practical S_i''
Clearance volume: theoretical S_0, practical S_0''

In the case of i/c engine driven compressors, specific energy consumption is calculated in terms of grams of fuel per cubic metre (g/m^3).

Reciprocating Compressor Performance

With a reciprocating compressor the capacity is not affected by the working pressure, unless this affects internal leakage. The idealized compressor diagram then takes the form shown in Fig 4.

The displacement of a reciprocating compressor is expressed as the net volume actually displaced by the piston at rated machine speed or unit of time, or that of the first stage in the case of multi-stage machines. The capacity is defined as the actual volume rate of flow of gas compressed and delivered at the standard discharge point, referred to conditions of total temperature, total pressure and composition (*eg* humidity) prevailing at the standard inlet point.

The practical form of the volumetric efficiency formula is:-

$$\eta_V = K\left[\left(r^{\frac{1}{n}} - 1\right) \times (C - A)\right]$$

where

r = compression ratio
n = polytropic exponent
C = cylinder clearance expressed as a decimal
A = correction factor for leakage, *etc*, typical values being 0.025–0.035 for lubricated machines to 0.07 for unlubricated machines.

The factor K is an empirical value of less than unity, allowing for other practical losses such as incomplete filling of the cylinder on each stroke. A typical value is 0.95.

$$\text{power required} \propto PV\left(\frac{n}{n-1}\right)\left(r^{\frac{n-1}{n}}-1\right)$$

where

P = cylinder inlet pressure
V = volume flow at cylinder intake conditions
r = compression ratio

In practical units:

$$\text{kW required} = 0.0024\,PV\left(\frac{n}{n-1}\right)\left(r^{\frac{n-1}{n}}-1\right)$$

where

P is in bar
V is in m^3/min

In practice the actual power required is normally determined from an indicator diagram which will indicate the thermodynamic reversible adiabatic process, which can be generally classified as cylinder losses. Further losses can result from fluid friction through inlet and discharge valves, together with mechanical (friction) losses. In general, with good design and construction, compression efficiency can be expected to be of the order of 85 to 95%, and mechanical efficiency from 83 to 93% is typical. This means that overall efficiency is likely to be between 75 and 85%, compared with ideal isentropic compression.

Power saving is possible with multi staging. Increasing the number of stages normally increases the overall efficiency and thus the economy of operation of the machine. It must be appreciated in such cases, however, that cylinder diameters are usually selected for specific intake and discharge pressures and operational pressures other than design values can have a marked effect on the performance of these machines.

Other Positive Displacement Types

Other types of positive displacement compressors — *eg* rotary screw and vane compressors — may have a fixed pressure ratio. The form of the compressor diagram is similar, but if they are operated above or below the design pressure, the diagram is modified as shown in Fig 5.

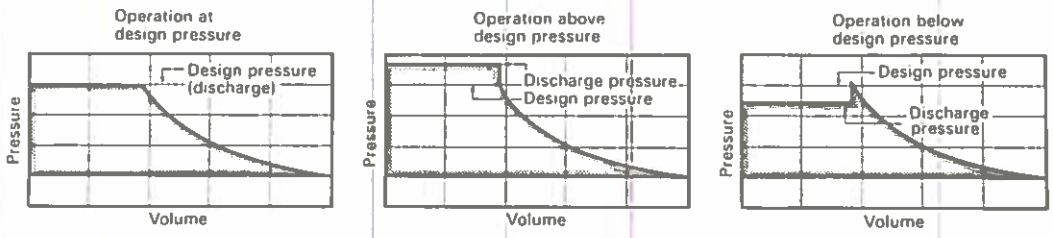


Fig 5 Operation at pressures deviating from the built-in pressure ratio.

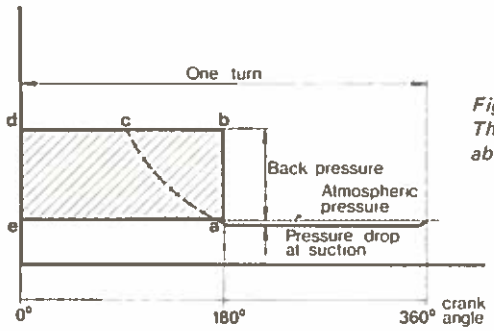


Fig 6 The working diagram of a Roots' blower. The compression work is represented by the area abdea. The area acdea represents the work required for compression in a piston type compressor. (Atlas Copco).

The typical diagram for a lobe rotor compressor is different again — Fig 6.

Dynamic Compressors

Any turbo-machine operated at constant speed is capable of delivering a wide range of volumes with only a small variation in pressure. If volume is decreased by throttling on the discharge side, pressure will also decrease. Pressure developed when the discharge is completely throttled will depend on the machine design and operating speed, but will always be lower than the maximum pressure. It is thus a characteristic of aerodynamic machines that dangerously high pressure cannot be built up under any operating conditions and so relief valves are not necessary in the discharge line. Regulation can be achieved either by throttling on the discharge side or variation in the impeller speed (or both).

The use of a valve offers the simplest form of control over pressure or capacity but any degree of throttling represents an irrecoverable power loss. Speed variation is, therefore, a more economic form of control when the compressor characteristics can be expressed in the form of a series of curves which may be superimposed on the demand curve to establish the required operating point — Fig 7. In other words, by adjusting the speed, the compressor characteristics may be matched to any required values of pressure and capacity within the operating limits of the compressor and its driver. Efficiency will not be at a maximum, however, unless this operating point corresponds to the design value for maximum efficiency.

The variable speed, capacity and adiabatic head can be related in terms of specific speed, viz:

$$\text{Specific speed } (n_s) = \frac{Q \times \text{rev/min}}{H_a^{0.75}}$$

where

Q = inlet capacity

H_a = adiabatic head per stage

This is a relative quantity only and has little significance other than as a design specific speed or value at which a given impeller in a given type of casing shows its maximum efficiency. It then serves as an index of classification for the form of the impeller (in that type of casing). In fact, the compressor may be operated over a range of speeds, capacities and adiabatic heads, and thus a wide range of specific speeds. These actual or operating specific speeds are useful only as a basis for comparing performance relative to the performance obtained at the design specific speed.

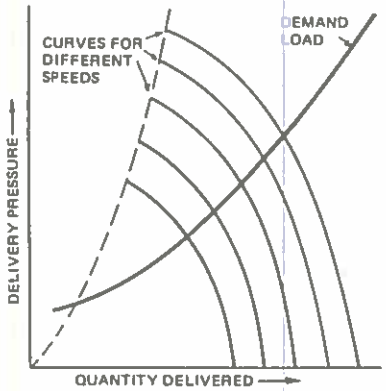


Fig 7

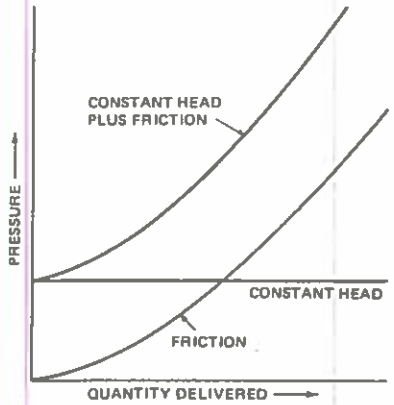


Fig 8

The demand load itself will depend on the operating conditions and may involve working against a constant head or pressure, against frictional resistance, or a combination of both. In the former case the demand load will be constant, and in the other two cases variable with capacity — Fig 8. The resultant demand curve may also vary under actual working conditions due to a change in head or frictional resistance. Equally the working point of the compressor can be adjusted by varying the demand curve, *eg* by means of a butterfly valve installed preferably near the compressor inlet or alternatively, near the discharge. Other methods of adjusting the compressor characteristics, and thus the pressure and capacity, include the use of adjustable diffuser vanes and adjustable inlet guide vanes.

Temperature Effects

A change in ambient air temperature will affect the intake volume required to produce a given volume of compressed air at a particular pressure. From the gas equation $PV = RT$ it follows that any volumetric change δV will be directly proportional to temperature change δT. Also, since power requirement is directly proportional to the volume of air inducted, this will be directly proportional to the temperature ratio.

Specifically this implies an advantage (power saving) if the ambient air temperature is lower than the 'design' air temperature at which the compressor performance is assessed; and a disadvantage (power increase) if the ambient temperature is higher.

Fig 8 illustrates this effect of intake air temperature on compressor performance, based on a design ambient air temperature of 15°C (60°F).

Discharge Temperature

The theoretical discharge temperature from a compressor is given by

$$T_d = T_i \times r^{\frac{n-1}{n}}$$

where

T_d = discharge temperature (absolute)
T_i = intake temperature (absolute)

For adiabatic compression (n = 1.4)

$$T_d = T_i r^{0.286}$$

where

r is the compression ratio

A more accurate formula is

$$T_d = T_i \left(\frac{T_i (r^{0.286} - 1)}{\eta} \right)$$

where

η = volumetric efficiency

In practice the discharge temperature will normally be reduced by cooling applied to the compressor and thus in practical systems may be of a similar order to the ambient temperature. Typical discharge temperatures (without cooling) relative to an initial intake temperature of 15°C (60°F) are given in Table I.

TABLE I – DISCHARGE TEMPERATURES

(Intake Temperature 15°C (60°F) – Intercooling to 15°C (60°F)

Pressure		Single Stage		Two-Stage		Three-Stage	
bar	lb/in^2	Temperature °C	Temperature °F	Temperature °C	Temperature °F	Temperature °C	Temperature °F
0.70	10	62	143	–	–	–	–
1.38	20	96	205	–	–	–	–
2.00	30	124	255	–	–	–	–
2.75	40	145	293	–	–	–	–
3.51	50	168	334	84	183	–	–
4.1	60	187	368	91	196	–	–
4.8	70	203	398	98	208	–	–
5.5	80	219	426	103	218	–	–
6.2	90	233	452	109	228	–	–
7.0	100	247	476	114	237	–	–
7.5	110	260	499	119	246	–	–
8.0	120	271	520	123	254	83	182
9.0	130	280	538	127	261	85	185
9.5	140	298	560	131	268	88	191
10.0	150	302	578	135	275	91	196
14.0	200	349	659	151	303	99	211
17.0	250	384	726	163	326	107	225
34.0	500			207	404	132	269

TABLE II – STANDARD ATMOSPHERE ACCORDING TO NASA

Altitude m	Pressure bar	Temperature °C	Density kg/m³
0	1.013	15.0	1.225
100	1.001	14.4	1.213
200	0.989	13.7	1.202
300	0.978	13.1	1.190
400	0.966	12.4	1.179
500	0.955	11.8	1.167
600	0.943	11.1	1.156
800	0.921	9.8	1.134
1000	0.899	8.5	1.112
1200	0.877	7.2	1.090
1400	0.856	5.9	1.069
1600	0.835	4.6	1.048
1800	0.815	3.3	1.027
2000	0.795	2.0	1.007
2200	0.775	0.7	0.986
2400	0.755	−0.6	0.966
2600	0.737	−1.9	0.947
2800	0.719	−3.2	0.928
3000	0.701	−4.5	0.909
3200	0.683	−5.8	0.891
3400	0.666	−7.1	0.872
3600	0.649	−8.4	0.854
3800	0.633	−9.7	0.837
4000	0.616	−11.0	0.819
5000	0.540	−17.5	0.736
6000	0.472	−24.0	0.660
7000	0.411	−30.5	0.590
8000	0.356	−37.0	0.525

TABLE III – INPUT POWER REQUIRED BY COMPRESSOR IN RELATION TO ALTITUDE (kW*s/litre)

Altitude	Single Stage		Two Stage	
	bar–gauge		bar–gauge	
Metres	4	7	4	7
0	0.258	0.349	0.232	0.302
300	0.254	0.342	0.229	0.296
600	0.251	0.336	0.226	0.290
900	0.248	0.330	0.221	0.284
1200	0.243	0.325	0.218	0.280
1500	0.237	0.320	0.214	0.273
1800	0.232	0.316	0.210	0.268
2100	0.229	0.309	0.205	0.262
2400	0.226	0.304	0.200	0.256
2700	0.222	0.298	0.197	0.251
3000	0.215	0.294	0.194	0.246
3600	0.206	0.283	0.186	0.237
4200		0.272	0.178	0.226

Effect of Altitude

Ambient pressure and temperature both decrease with altitude, as given in Table II. These changes affect the pressure ratio of compressors (and air tools), which can appreciably modify performance. The extent to which this is realized will depend to some extent on the type and design of compressor, but the following are typical percentage reductions in capacity and power consumption for two-stage compressors working at 8 bars.

Compressor Type	% Reduction for each 1000 metres increase in altitude	
Piston Type	Capacity	Power Consumption
Medium size, air cooled	2.1	7.0
Large piston type, water cooled	1.5	6.2
Screw, oil injected	0.6	5.0
Large screw, water cooled	0.3	7.0

It should be noted that increasing altitude will also have an effect on the performance of internal combustion engines driving compressors, usually roughly equivalent to a 1% loss of power for each 100 m of altitude (which can be halved by turbo-supercharging) — see also Table III. The actual effect of high ambient temperature is rather more marked — typically a 4% power loss for every 10°C rise in ambient temperature. This cannot be offset by turbo-supercharging. The performance of electric motors is not affected by altitude, but the lower density of air will result in poorer cooling. To offset this it may be necessary to reduce the shaft power at altitudes above 1 500 m. Electric motor shaft power should also be reduced by 5% for every 5°C ambient temperature rise above 40°C, again from cooling considerations.

Compressor Noise Suppresssion

TYPICAL SOURCES of compressor noise and methods of treatment are summarized in Table I. It should also be emphasized that compressor sound levels measured under free field conditions may be considerably modified by the room characteristics when installed; backyard noise levels present; and the length of suction and discharge connections and type of connection.

Provided the sound emanates mainly from the compressor unit, acoustic enclosure of the unit is normally the most satisfactory treatment. In this case the significance of small leaks or openings in the enclosure cannot be over-emphasized. They provide 'escape paths' for noise which can multiply the absorption otherwise expected from a well designed enclosure.

Some cases do not justify complete enclosure; or such treatment may be impractical (*eg* with a portable compressor on site). In this case a sound barrier may provide a satisfactory answer, sited to reduce noise transmission in the required direction.

The degree (and type) of treatment necessary also depends on the type of compressor and, in the case of reciprocating compressors particularly, degree of balance of reciprocating masses. Rotary compressors can be much more refined as regards dynamic balance, reducing vibration levels, and are inherently quicker than reciprocating machines.

Stationary Compressors

The following summarizes the main treatments to be considered in reducing noise levels with stationary compressors:

(i) Complete enclosure of the compressor and driver in a properly soundproofed room or chamber. Isolation treatment may also have to be applied to ensure that no vibrations are transmitted through the walls or floor.

(ii) Choice of a rotary compressor, which is inherently quieter than a reciprocating machine. The rotary compressor runs with absence of the vibration and freedom from the pulsations inherent in reciprocating machines. Provided effective intake silencing is employed, the bulk of the noise will be that generated by the driver, which should be amenable to direct treatment.

(iii) Good alignment and rigidity of coupling without transmitting vibrations between driver and compressor are also important factors in noise control.

TABLE I – COMPRESSOR NOISE : SOURCES AND TREATMENTS

Source	Treatment
1. Driver vibration	Isolation mounting
2. Compressor vibration	(i) isolation mounting (ii) decoupling
3. Driver running noise	(i) intake silencing (ii) exhaust silencing (iii) enclosure
4. Compressor running noise	(i) intake silencing (ii) enclosure
5. Cooling fan*	Fan should be designed for minimum noise
6. Gear noise	Complete enclosure
7. Relief valve operation	Complete enclosure
8. Resonance – whole chassis* canopy*	Isolation mounting and decoupling Isolation Suitable damping and isolation mounting

*Applicable to portable compressors

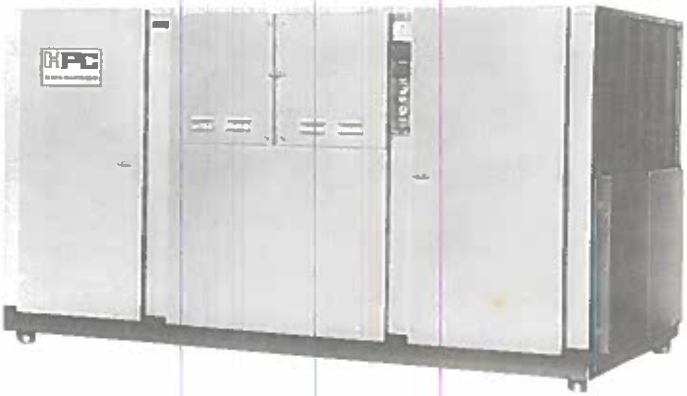
Pipe noise should also be considered. Noise radiated from pipes can, in fact, exceed that radiated by the compressor and driver. If it only equals that of compressor/driver noise, complete suppression of the latter would only reduce the total noise level by 3 dB.

Small diameter heavy walled pipes do not normally need treatment, other than the use of isolating points to eliminate transmission of vibration to adjacent structures. The sizes of piping associated with larger compressors, however, require proper lagging with sound absorbing material and a dense cover or jacket.

A number of other points affecting design in relation to noise suppression can be considered. There is, for example, the use of an oil sump within the compressor housing – a feature which further helps to muffle the noise. In the cooling area, attention to fan design, air speeds, and construction of the after-cooler all minimize noise. Air intake is often a source of considerable noise and must be controlled. This means that attention has to be given to the pattern of air flow, its velocity and the position of the air filter in the system.

Portable Compressors

Mobile compressors are commonly designed as truck or trolley mounted units with a diesel engine driver. Smaller portable compressors may have spark-ignition drivers (or possible electric motors).



HPC 110 kW (150 horsepower) screw compressor is mounted on rigid chassis with removable noise-reducing enclosure panels.

The modern approach to silencing a portable compressor is to identify each noise source systematically and to reduce its level. The total sum of all noise levels does not add up, however, to the total annoyance factor to the human ear. There are some low frequencies whose annoyance factor is out of all proportion to their magnitude. Using the right shape and size of canopy has the effect of reducing resonance and hence these objectionable vibrations.

The ducting required on a portable compressor to allow the free passage of cooling air conflicts with the silencing concept, because air passing through each duct opening is a potential noise source; hence the need to consider the total noise source and to balance the outcome.

Whilst portable compressors are silenced to 75 dB(A) at 7 metres (23 feet) with little extra cost, silencing thereafter becomes progressively more expensive, but generally feasible.

More and more attention is being given to producing mobile compressors for road breaking operations, *etc*, where the noise level conforms to the following levels:

70 dB(A) in rural, suburban and urban areas away from main traffic and industrial noise.

75 dB(A) in urban areas near main roads in heavy industrial areas.

Rather different conditions apply in the case of stationary compressors in industrial applications where each case normally has to be considered on its own merits. Complete enclosure is usually the best answer where appreciable reduction in noise has to be realized.

See also chapter on *Noise Control*.

Compressor Lubrication

LUBRICATION IS an extremely important parameter in compressor performance, its function being to dissipate frictional heat, reduce wear on sliding and stationary surfaces, reduce internal leakage and protect parts from corrosion. In addition the lubricant can serve the duty of flushing away wear products, defects and moisture.

Both the choice of lubricating system and the choice of lubricant are important, particularly as lubrication requirements differ with different types of compressor and are in different sections of the machine in the case of reciprocating compressors.

TABLE I – GENERAL GUIDE TO COMPRESSOR LUBRICATION

Type of Compressor	Type of Oil	Typical Oil Viscosity*
Air-cooled reciprocating up to 20 kW	Single-grade† engine oils to API classifications SD, SE, CC or CD; or industrial compressor oils	SAR 10W, 20/20W or SAE 30
over 20 kW	Single-grade† engine oils to API classification CC or CD; or industrial compressor oils	ISO VG68 or 100 SAE 20/20W or SAE 30
Water-cooled reciprocating	Industrial compressor oils	ISO VG68 or 100
Oil-flooded rotary screw compressors	Special rotary compressor oils or inhibited turbine oils	ISO VG32
Sliding vane	Industrial compressor oils	ISO VG150 or 220
'Oil-free' compressors	Industrial compressor oils, turbine oils or hydraulic oils	ISO VG68 or SAE 20/20W

*As recommended by compressor manufacturer † Multigrade oils are not suitable

Note: As a general rule synthetic lubricants should not be used in compressors unless the compressor manufacturer approves and/or recommends these as satisfactory.

Lubricants

The use of synthetic oil lubricants for compressors is increasing, being mainly attractive because of their low deposit formation with long life and good fire-resistant properties. They are more expensive than mineral oil lubricants, can attack paints and may call for special gaskets and seal materials. On the other hand, oil carry-over is reduced with synthetic lubricants, so compatibility problems will not necessarily occur.

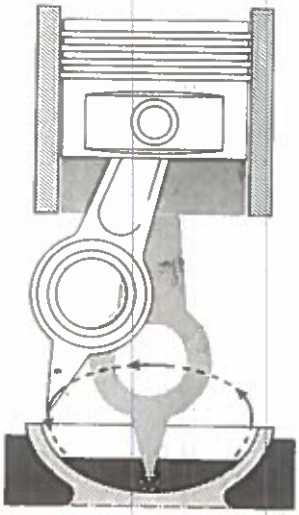
Mineral oils remain the most common type in use, straight cut naphthenic oils of medium to high viscosity normally being preferred. Compressor manufacturers normally specify grades for use in their compressors — see also Table I.

Lubrication Systems

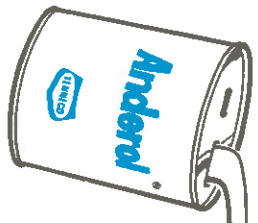
Experience has shown that certain lubrication systems are best for specific types of compressors, although there are normally alternative choices available to the designer. The most common systems used are:-

(i) splash lubrication
(ii) flood lubrication (gravity lubrication)
(iii) force-feed lubrication
(iv) injection lubrication

In the case of reciprocating compressors the simplest arrangement is *splash lubrication*, whereby a projection on the connecting rod big end dips into oil held in the crankcase. The crankcase thus acts as an oil reservoir and must be closed with the exception of a small breather hole. Lubricant is dispersed mainly in the form of an oil mist reaching the bearings, surfaces and cylinders. For best control the splash oil should be taken from a trough. Failing this, some provision should be made to maintain the crankcase oil at a constant level. Water which collects in the bottom of the crankcase should be drawn off at regular intervals.



Splash lubrication with controlled feed. (Atlas Copco)



One manufacturer saved £10,000 in one year, using Anderol® Synthetic Lubricants in his air compressors

...that's just one story

Wherever ANDEROL diester-based synthetic lubricants are used, the story is the same: less wear and tear on moving parts, drastically reduced carbon and sludge build up, extended oil change intervals, reduced make up oil and expensive down time. ANDEROL also features low toxicity, biodegradability, high film strength, and 100°F higher auto ignition points than petroleum oils.

There is an ANDEROL synthetic lubricant for almost every use:

compressor oils for air and process gas applications, impregnation fluids, instrument and watch oil, office equipment lubricants, gun lubricants, greases, and the new ANDEROL 800 engine oil for diesel and petrol use. Each designed to cut the cost of maintenance by extending parts life and service intervals.

Manufactured by: TENNECO®

ANDEROL SYNTHETIC LUBRICANTS
Tenneco Colortrend B.V.
A Tenneco Chemicals Europe Company

Sole U.K. Agents and distributors:

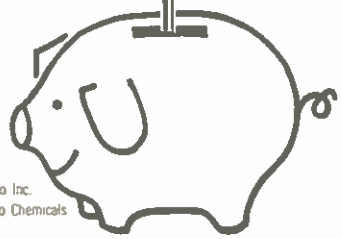
LAWRENCE INDUSTRIES

Mitcham Industrial Estate,
Streatham Road, Mitcham,
Surrey CR4 2AP
Telephone: 01-648 2272
Telex: 946846

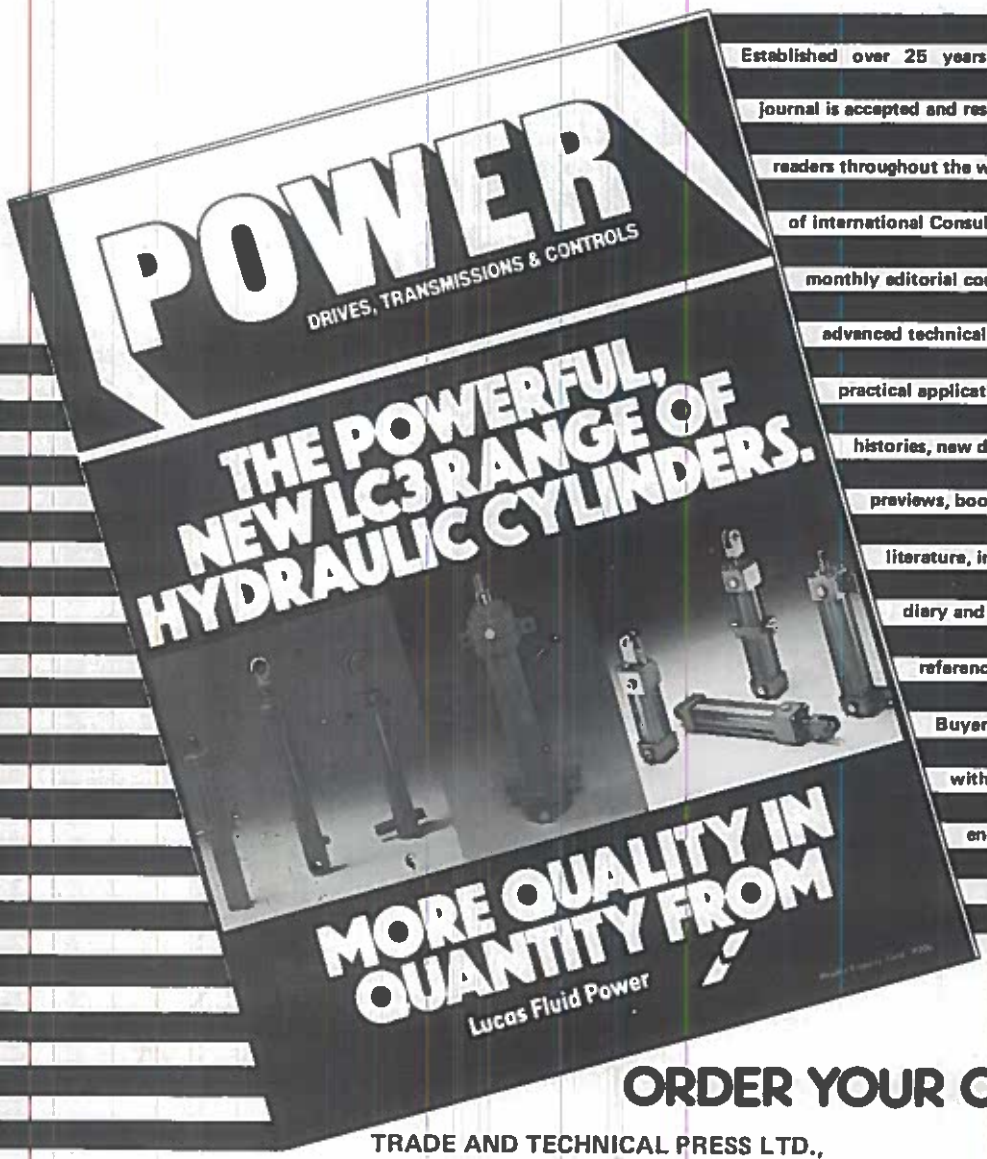
FIND OUT MORE ABOUT THE LUBRICANTS OF TOMORROW TODAY

Name _____
Company _____
Address _____
_____
Application _____

® Reg. T.M. of Tenneco Inc.
® Reg. T.M. of Tenneco Chemicals



THE international magazine for Power Transmission Engineering !


POWER
DRIVES, TRANSMISSIONS & CONTROLS
THE POWERFUL, NEW LC3 RANGE OF HYDRAULIC CYLINDERS.
MORE QUALITY IN QUANTITY FROM
Lucas Fluid Power

Established over 25 years, this monthly technical journal is accepted and respected by thousands of readers throughout the world. With a panel of international Consultant Editors, monthly editorial content includes: advanced technical articles, practical applications, case histories, new developments, previews, book review, literature, international diary and a ready reference fold-out Buyers' Guide with reader enquiry cards.

ORDER YOUR COPY NOW!

TRADE AND TECHNICAL PRESS LTD.,
CROWN HOUSE, MORDEN, SURREY, SM4 5EW, ENGLAND.

ONLY BP GIVE YOU SUCH A BIG HELPING HAND.

Being Britain's biggest company naturally has its advantages.

Advantages that we're happy to hand on to you.

Take our international research and development facilities for example.

They are among the most impressive in Britain, employing some of the country's finest scientists.

Which means we've a lot of technical knowledge and expertise in the palm of our hand.

But we're far from tight-fisted with our technology.

BP's new range of Dielectric Fluids, for example, are EDM oils which deal with machining problems varying from precision work to heavy duty roughing.

But they also look after the people who work with them. They have high flash points and low levels of aromatic hydrocarbons. They resist oxidation and keep fuming and unpleasant odours to a minimum. All without sacrificing an exceptionally long working-life helped by excellent filterability.

They possess high dielectric strength, low conductivity and permit maximum metal removal.

Evidence that, yet again, BP are on hand to help get the job done.

For further information about the new BP Dielectric Fluids telephone 01-821 2756 or send this coupon to Room 0401, BP Oil Limited, BP House, Victoria Street, London SW1E 5NJ.

Name _____

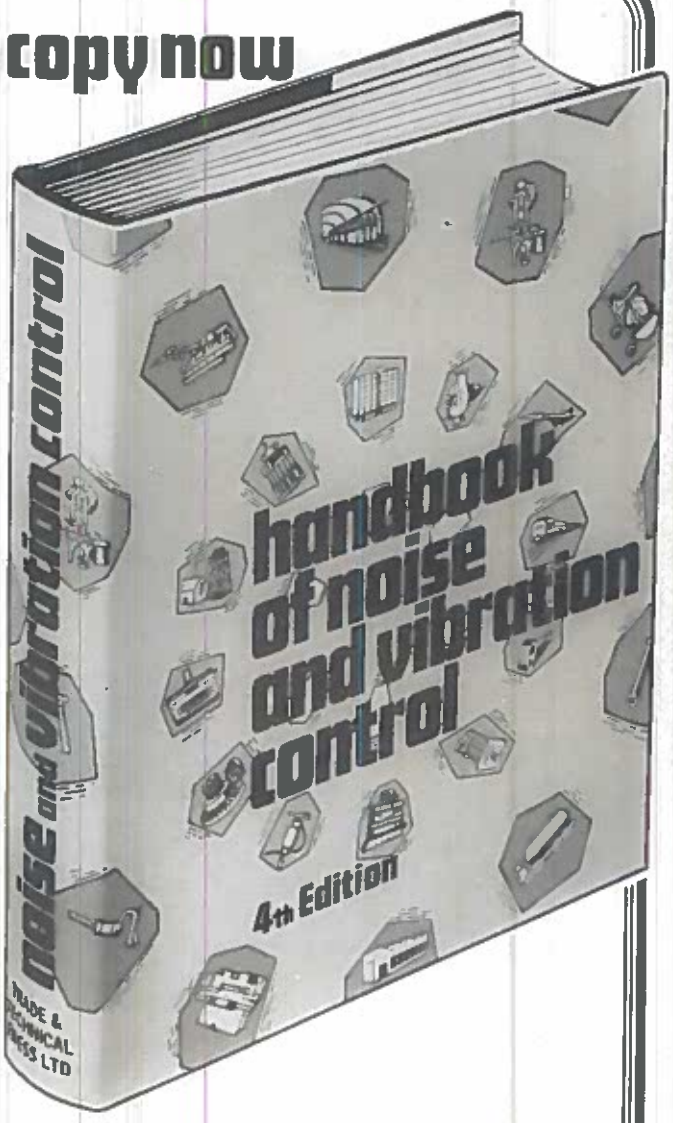
Address _____

_____

Tel. No. _____ HH/PH1

ANOTHER HELPING HAND FROM BP

order your copy now



- Completely revised and up-dated.
- Handbook of vital importance to industry and commerce.
- Over 800 pages, 2,500 illustrations, diagrams, charts, tables, etc.

THE 4TH EDITION of this important reference work is the most practical and useful handbook available for engineers, designers, architects, R. & D. personnel.
Contents embrace: Section 1: Sound and Noise Parameters, Sound Physics, Propagation of Sound, Measurement of Sound, Physchoacoustics, Industrial Hearing Conservation, Speech Communication, Vibration and Shock. Section 2: Machine Noise, Internal Combustion Engines, Electric Motors, Heating and Air Conditioning Plant, Construction Site Equipment, Pumps and Pumping Systems, Compressed Air Noise, Fan Noise, Hydraulic Systems. Section 2B: Sound Level Meters, Noise Measurement Techniques, Analysing Noise, Environmental, Noise Monitoring, Audiometry, Vibration Measuring Instruments, Vibration Analysis, Dynamic Analysis, Real Time Frequency Processing, Modal Analysis, Digitial Analysis, Machinery Health Monitoring, Vibration Generators. Section 3A: Noise Control Criteria, Sound Barrier, Semi-Enclosures and Screens, Sound Absorption, Sound Insulation, Sound Treatment Materials, Partitions and Panels, Enclosures, Acoustic Treatment with Sheet Lead, Industrial Silencers, Automotive-type Silencers, Windows, Personal Hearing Protection. Section 3B: Vibration Isolation Criteria, Damping, Vibration and Balance in Machines, Anti-Vibration Mounts, Shock Absorbers and Dampers, Elastomeric Bearings. Section 3C: Building Noise Criteria, Factories and Workshops, Offices and Commercial Buildings, Public Buildings, Domestic Buildings, Resilient Mounting of Structures, Auditoria. Section 3D: Noise and the Environment, Aircraft Noise, Road Traffic Noise, Road Vehicle Noise, Railway Noise, Hovercraft Noise, Helicopter Noise, Discotheques Noise. Section 3E: Legal Aspects and Legislation, Codes, Standards and Recommendations.

TRADE & TECHNICAL PRESS LTD., CROWN HOUSE, MORDEN, SURREY SM4 5EW, ENGLAND.

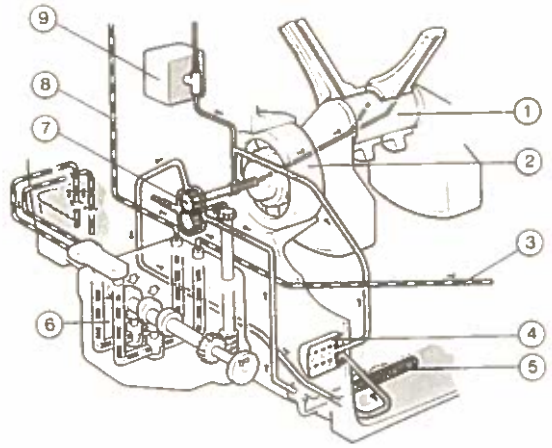
Splash lubrication is largely limited to small single-acting reciprocating compressors. It has a number of inherent disadvantages, notably:-

(i) the lubricant cannot be filtered and lubricant life is limited

(ii) lubrication during starting up may be marginal

(iii) the bearing film of lubricant is thin and is not supplied under pressure.

Flood lubrication is widely used on horizontal compressors for lubricating bearings and crosshead. In this case the oil flow is under gravity after first being elevated by a disc, oil ring, chain or collar on the crankshaft which lifts it to a trough with distribution tubes or to the upper part of the journal(s). Basically, this provides better control of the amount of oil distributed than splash lubrication but the pressure developed is very low and not generally sufficient for a high speed compressor.



1 — Crankshaft with oil-channels
2 — Crankshaft roller bearing
3 — Oil pipe to LP-cylinder
4 — Oil-filter cartridge
5 — Oil sieve in crankcase
6 — Plunge-type lubricator for cylinder lubrication
7 — Gear-type oil pump for connecting rods and crosshead guides
8 — Oil pipe to HP-cylinder
9 — Oil-pressure switch

Force-feed lubrication. (Atlas Copco)

Force-feed lubrication utilizing a circulatory system is normally to be preferred. Feed is taken from an oil reservoir by a plunger or gear pump. Actual circulation may be by gravity, when the oil is pumped to an overhead reservoir and the circulation system is slightly more pressurized. Alternatively, the oil may be fed directly from the pump under pressure through pipes to the various lubricating points. Oil pressure is kept constant by a spring-loaded overflow valve. For cylinder lubrication of crosshead compressors special piston-type lubricators are normally used.

Sight-feed drip oilers are occasionally used to lubricate small parts. These are similar to cylinder lubricators and can be regulated within close limits provided they are kept in good condition. Siphon wick oilers may also be used when it is important that there should be no undue variation in oil level in the container. Grease lubrication may be provided at certain points (*eg* to lubricate rolling bearings). In this case the lubricating points are either fitted with grease nipples or screw-down grease cups.

In the case of cylinders the ideal oil feed is that which will provide efficient and effective lubrication with the minimum amount of oil, at the same time maintaining a perfect piston seal — an excess of oil is undesirable. A practical check is to service the discharge valves periodically and examine their appearance. They should not be dry or show signs of rust (insufficient oil) or be

excessively wet (excess of oil). They should have a wet appearance and be oily to the touch. Excess of oil can also be noted by the appearance of pools of oil lying in the cylinders, or surplus oil on the connecting rods.

Vane Compressors

Adequate lubrication is necessary in the case of vane-type compressors to minimize friction and wear on the vanes and/or wear ring and to assist in sealing. Quite high bearing pressures may be exerted by the leading edges of the vanes at the trailing edges of the slots, calling for a lubricant which forms a tenacious oil film. Bearing lubrication is normally separate, preferably by a force-feed lubricator although this is not usually allied to a circulatory system.

Oil for lubricating the vanes may be fed to the casing via the air intake, direct to the casing or to the rotor through its shaft. The first method is more usual, loading the intake air with oil by an atomizer. (Discharge air may be used to inject the oil into the intake). The other two methods involve force-feeding, either direct to the ends of the rotor or through a hole along the centre of the shaft. In the latter case interconnecting radial holes drilled along the length of the shaft or at the centre of the shaft throw out oil under centrifugal force.

Rotary Screw Compressors

Rotary screw compressors of the 'dry' type do not require internal lubrication since the rotors operate with positive clearance. Only shaft bearings and timing gears thus need lubrication. Plain bearings can be lubricated by ring oilers and timing gears by splash lubrication. Where rolling bearings are used, bearings at the timing gear end may be lubricated by oil and those at the other end by grease.

With oil-flooded screw compressors, oil is injected into the casing to provide both sealing and lubrication, requiring the use of a comprehensive oil recirculation system, including filters and oil cooler. Actual oil injection is normally accomplished using discharge air pressure, or may be direct from a separate pump.

Dynamic Compressors

Dynamic compressors work with running clearance between the rotor and casing; thus only the bearings need lubrication. Bearings are normally accommodated in sealed housings outside the main compressor casing and may be lubricated by ring oilers, force-feed (particularly for larger high-speed compressors), or grease.

Compressor Controls

SMALLER STATIONARY compressors may be run continuously with only a blow-off (pressure-relief) valve on the delivery side which opens when a specified maximum pressure is reached to allow compressed air to be blown off to atmosphere instead of being delivered to the system — Fig 1. Where the system incorporates a receiver, however, a check valve may also be incorporated ahead of the receiver to prevent high pessure being blown back into the compressor (Fig 2). This is strictly necessary with types of compressors which are valveless (*eg* vane compressors). It is not necessary with compressors with valves (*eg* piston and diaphragm types), but may still be used with them as a redundant backflow protective device. A safety valve may also be incorporated in the delivery side of the receiver to protect the system against over-pressure.

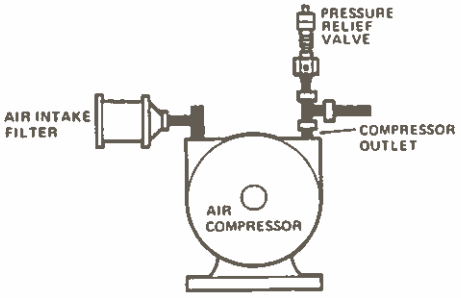


Fig 1 Basic low pressure power supply system.

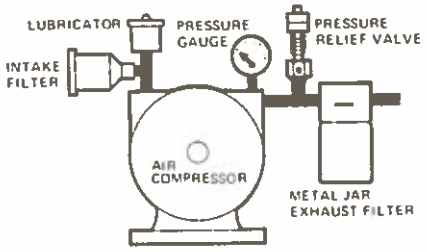


Fig 2 Pressure relief valve adds basic control capability to simple system.

For continuous duty applications it may be desirable to maintain a higher pressure in the receiver than that required in the supply line. In this case the receiver is followed by a pressure regulator — Fig 3.

An alternative method of pressure regulation is by on-off cycling using an electrical pressure switch on the receiver to cut the motor supply off at a predetermined 'high' pressure level, and back on again at a predetermined 'low' level — Fig 4. The receiver pressure is then held between these two pressure levels as long as demand does not exceed the recharging capacity of the compressor.

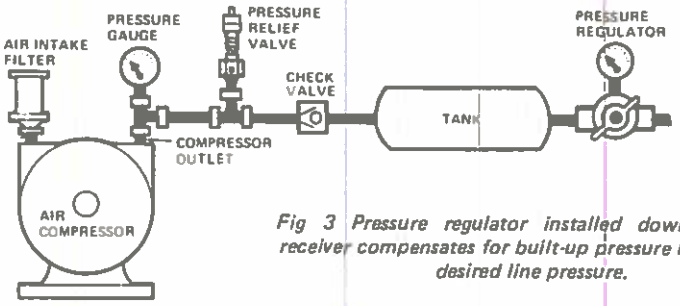
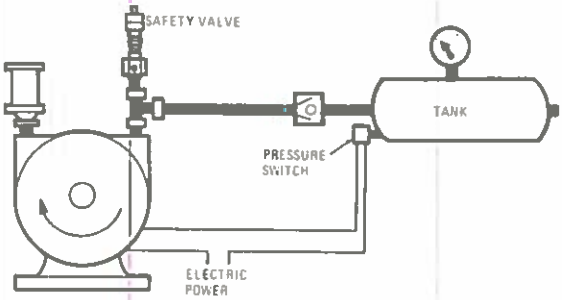
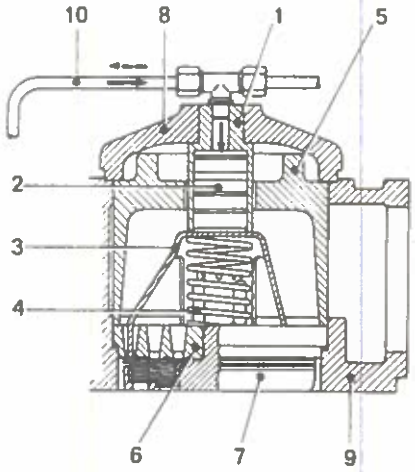


Fig 3 Pressure regulator installed downstream of receiver compensates for built-up pressure higher than desired line pressure.

Fig 4 A pressure switch controls receiver pressure independent of downstream pressure by on/off cycling of compressor drive motor.





1 – Unloading cylinder
2 – Unloading piston
3 – Unloading claw (depresses the valve discs and keeps the suction valve open during the unloading period)
4 – Unloading spring (returns unloading claw at start of loading)
5 – Valve retainer
6 – Seat of suction valve
7 – Suction valve guard
8 – Suction valve cover
9 – Cylinder head
10 – Compressed air to and from unloading device.

Fig 5 Unloading device for piston compressors

Unloading

Larger systems are commonly operated with the compressor running continuously with compression capacity regulated by *unloading*. One mechanical method used with reciprocating compressors is shown in Fig 5.

Pressure from the receiver is transmitted to the underside of the piston of the device, which acts against a spring set at a predetermined pressure. When the pressure in the receiver exceeds this figure, the spring compresses and allows air to flow to a piston on top of the suction valves, which in turn depresses a claw which holds the suction valves off their seat, so that compression cannot take place in the cylinder. This is termed *suction valve unloading.*

When double-acting compressors are used, by having two air relays set at different pressures, it is possible to unload each side of the piston in turn, giving full load, half load, and zero load. This is termed *three step unloading.*

A variation on this method is to allow the inlet valves to open normally to fill the cylinders but keep them open for a timed interval after the compression stroke. This can provide stepless regulation but is normally only used with process compressors.

A further variation is called *clearance pocket unloading.* Here the pockets are brought into communication with the cylinder by valves which enlarge the clearance volume and so reduce the output of the compressor. A normal method on double-acting compressors is to have a combination of clearance pocket and suction valve unloaders to give 100%—75%—50%—0% unloading. Clearance pocket unloading on its own can provide stepless regulation.

Reverse Flow Regulation

A more recent method of stepless control is *reverse flow regulation,* causing the intake to vary directly according to requirements by delaying the closing of the suction valves. This delayed closing of the suction valves is achieved by means of special unloaders. These are actuated by the infinitely variable pressure of a piston built onto the valve cover (usually a diaphragm piston in a servo-cylinder). Thus, a number of unloader springs are tensioned by means of an unloader plate, which in turn keeps the valve plate of the regulated suction valve open against the reverse flow of the gas during the compression stroke in the cylinder. Part of the gas taken in flows back into the suction duct and reduces the output capacity by an equal amount. During the compression stroke, the velocity of the gas flowing back rises steadily with increasing piston speeds. The flow forces acting on the unloader and valve plate in a closing action thus grow steadily with increasing piston speed (depending on the piston position) and eventually exceed the holding-open force which has been adjusted to a certain value. This causes the valve to close after a corresponding delay. The reverse flow of the gas is thus terminated. The remaining gas in the cylinder is compressed and discharged in the usual manner.

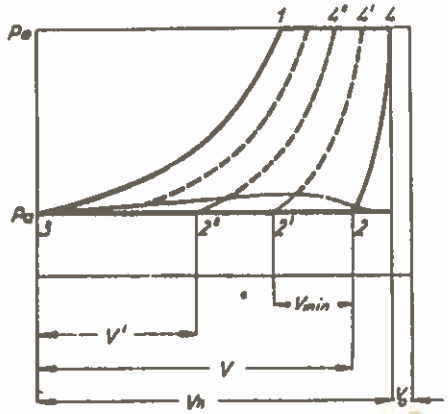


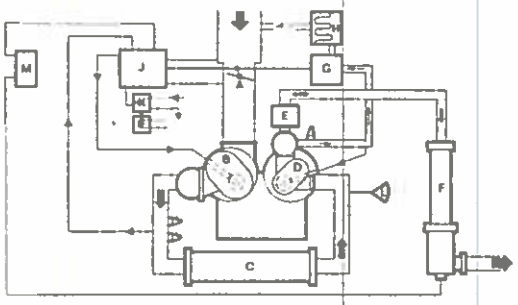
Fig 6 Regulating diagram of a reverse flow regulation (theoretical). (Hoerbiger)

This permits adjustment to any desired capacity between 100% and 40% of full load (see Fig 6), and in special cases stepless capacity reductions down to 15% and less may be achieved.

A further method of controlling the output is by throttling the intake, but this method requires more hp at part load conditions than the methods described earlier.

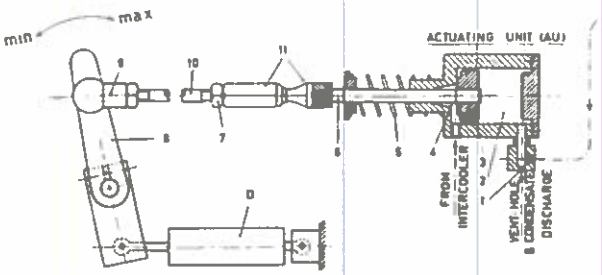
Screw Compressors

A different form of unloading is adopted in the case of screw compressors. The capacity is regulated by throttling the air inlet and, at the same time, opening a blow-off valve. The pressure side is shut off with a non-return valve. Under load the blow-off valve is closed and the throttle valve fully open. The unloading device is operated by a spring loaded automatic air relay of the same type as that used on reciprocating compressor controls. A complete installation is shown in Fig 7.



A – Throttle valve
B – Low-pressure stage
C – Intercooler
D – High pressure stage
E – Non-return valve
F – Aftercooler
G – Blow-off valve
H – Blow-off cooler
J – Servo valve
K – Solenoid valve
L – Temperature switch
M – Regulator

Fig 7 Capacity unloading of screw compressors. (Atlas Copco)



D – Damper, engine speed control lever
1 – Air vent hole and condensate discharge
2 – Chamber, converted air receiver pressure
3 – Piston
4 – Chamber, constant intercooler pressure
5 – Return spring, piston
6 – Piston rod
7 – Check nuts, ball joints
8 – Engine speed control lever
9 – Ball joint
10 – Operating rod, actuating unit
11 – Ball-and-socket joint

Fig 8 Speed regulating system for i/c engine compressor driver. (Atlas Copco).

Monitoring and Regulation

Compressor controls may also be allied to monitoring devices, or monitoring systems, capable of providing automatic shut-down in the event of failure of the system approaching a dangerous condition. The number of monitoring devices required varies with the type and complexity of the system, and the exact number can be decided according to need.

Continuous regulation is widely applied to i/c engine driven compressors, particularly portable compressors. The principle employed is of speed regulation of the driver to obtain continuous control of the compressor pressure – *eg* by means of a slide valve sensing the final pressure and controlling the position of the throttle valve – see Fig 8.

The first practical speed regulating control system of this type coupled running speed to air demand, and allowed the engine to idle when the demand for air was zero, thus reducing wear and tear on both engine and compressor and also reducing overall fuel consumption. This was followed by stepped controls enabling the engine speed and compressor speed to vary exactly, according to demand. This has subsequently become a more or less standard feature on all larger sizes of portable reciprocating compressors, although there are difficulties in adapting a similar form of control to rotary machines.

Stop-Start Controls

Stop-start control may also be combined with constant speed control in a *dual control* system, when either type of control can be selected as required. *Automatic dual control* is a refinement whereby air demand is sensed and the control automatically selects the form of operation required – *ie* constant speed or stop-start.

Compressor Protection

Critical parameters which need to be maintained to provide basic protection for a compressor are coolant water flow or temperature, discharge air temperature, and lubricating oil pressure or level (in the case of lubricated compressors). Typical minimum parameters, *ie* danger levels, might be

(i) coolant water flow 2/3 of normal flow rate

(ii) discharge air temperature 160°C

(iii) lubricating oil pressure 2 bar

Temperature sensors can provide cover for (i) and (ii), and pressure sensors for (iii). If in the form of temperature- or pressure-operated switches they can provide automatic protection either as alarm devices, or control devices (automatically shutting down the system when a critical parameter level is reached).

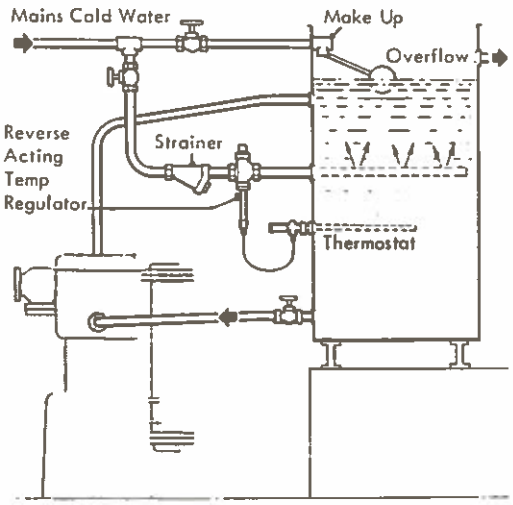


Fig 9 Temperature regulator controlling introduction of water to holding tank. (Sarco).

Where compressors employ closed-circuit water cooling (either thermo-syphon or forced circulation), additional cooling can be provided from the main or similar cold water supply via a temperature regulator. This allows cold 'make-up' water to be supplied to the cooling tank at a controlled rate, hot water being displaced through an overflow. A typical installation of this type is shown in Fig 9.

Automatic temperature control can also be applied to mains water cooling systems, the purpose in this case being both to eliminate wastage of water and to prevent excessive cooling which could cause condensation within the cylinder. A typical installation applied to a two-stage compressor is shown in Fig 10. It is important that the temperature regulator cannot shut off the supply of cooling water completely. This can be guarded against by incorporating an adjustable bleed in the valve itself (ensuring a small flow when the valve is shut); or the use of a bypass, the flow through which is controlled by a needle valve.

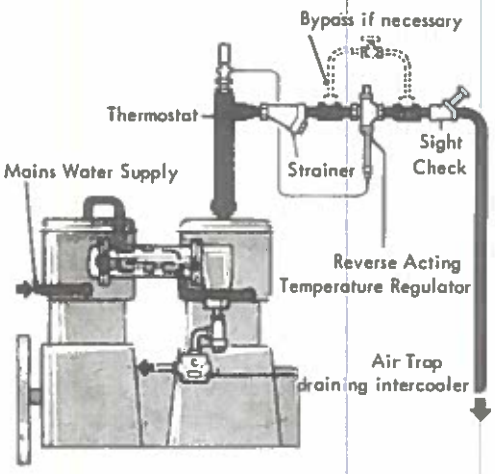
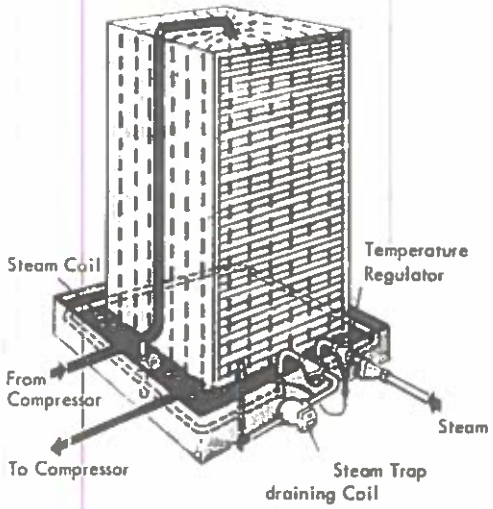


Fig 10 Temperature regulator controlling mains water cooling of two stage compressor. (Sarco).

Fig 11 Temperature regulator controlling temperature of cooling tower sump. (Sarco).



In the case of larger compressors operating with closed cycle, cooling is accomplished by passing the water through a mechanical cooler or a cooling tower. Again temperature control can prove advantageous, particularly to avoid too low a temperature being developed in the second stage cylinder.

An important consideration in the case of cooling towers is the possibility of freezing, with consequent cessation of water circulation calling for the compressor to be shut down. This possibility can be avoided by using a temperature regulator valve commanding a supply of steam to a heating coil in the sump. The regulator can be set to open when the water temperature falls below, say, 2°C (35°F), so that no steam is used until near-freezing conditions are approached — Fig 11. To guard against the possibility of the cooling tower freezing solid, the line from the diversion control valve can be valved to return water from the compressor direct to the sump instead of the top of the tower under low temperature conditions.

Temperature regulators can also be used to advantage on water cooled aftercoolers. The use of a temperature regulator enables the water consumption to be set to the point of maximum efficiency, and maximum water economy. An example of the use of a temperature regulator controlling main water supply to an aftercooler is shown in Fig 12.

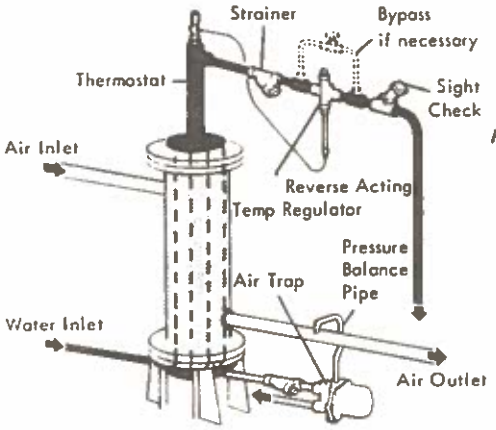


Fig 12 Temperature regulator controlling aftercooler. (Sarco).

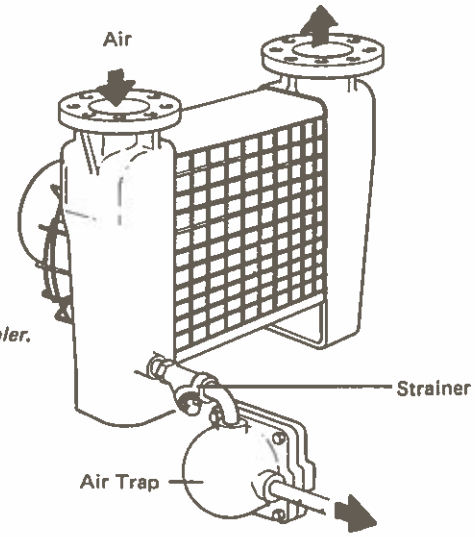


Fig 13 Air blast aftercooler. (Sarco)

Water cooled aftercoolers are usually the most efficient and the lower the air temperature they can produce the better. Air cooling has also come more to the fore in recent years, particularly in areas where cooling water is expensive or not readily available. An air-blast aftercooler is then usually the first choice. The principle on which these work is that an electric motor driven fan directs a stream of air over a bank of finned tubes through which the compressed air flows — *eg* see Fig 13.

SUMMARY OF COMPRESSOR CONTROLS

Type of Compressor	Suitable Control System(s)
Reciprocating	(i) stop/start — intermittent operation (ii) constant speed running with 'stepped' discharge achieved by: (a) inlet valve blocking, or (b) intake throttling, or (c) inlet valve unloading or clearance pocket, or (d) external bypass (iii) variable speed
Rotary screw	(i) intake throttling, or (ii) external bypass (iii) variable speed
Rotary vane	(i) stop/start — intermittent operation (ii) constant speed running with (a) intake throttling, or (b) external bypass (iii) variable speed
Dynamic machines	(i) constant speed running with intake throttling (ii) variable speed

Compressor Installation

A COMPRESSOR installation may be *centralized* where all the compressors needed are housed in a single compressor plant; or *decentralized* where individual compressors are located near the points of use. A *centralized* installation can show greater cost-effectiveness as well as making it easier to optimize intake conditions and silencing. Modern practice with centralized installations is to install individual compressors with their associated ancillary equipment as completely separate units. It is then possible to shut down a single compressor for maintenance without shutting down the entire installation.

Ideally a compressor should be located where it can induct clean, dry and cool air. Another primary consideration is that a compressor is inherently a noisy machine and, unless specifically designed or treated for noise control, should be located in a 'noisy' area of a factory or workshop. In the case of a centralized installation it is generally desirable (and virtually necessary in the case of larger machines) to have the compressor in a separate well ventilated room or chamber, inducting air from the outside of the building. Each compressor should have a separate air intake, and pressure drop over the intake system should be kept low.

Basic treatment which can be applied is the fitting of an air intake filter of a suitable type and size to ensure that air reaching the compressor is clean; and a separate air intake silencer to reduce 'intake noise', which is usually a predominant noise-generator. Further noise treatment may, or may not, have to be applied to the machine itself and the room (*eg* the provision of adequate ventilation can establish noise-escape paths which may have to be dealt with).

In the case of reciprocating compressors the intake flow is pulsatory. This can aggravate noise problems through resonance with critical lengths of intake piping as well as affecting flow into the compressor. As a general rule the chance of resonance occurring is greatest when the intake pipe length corresponds to a quarter or three-quarters of the pressure pulse wavelength (λ) which for a single-acting compressor is:

$$\lambda = \frac{61.5 \times T^{0.5}}{n}$$

where λ = wavelength in metres
T = absolute temperature, K
n = compressor shaft rev/min

Note: the quantity $(61.5 \times T^{0.5})$ is the velocity of sound in air in m/sec.

Compressor Intake

The intake system must be designed with low pressure drop in mind, implying a limit on intake air velocities. At the same time intake flow should be smooth rather than pulsating as this will appreciably reduce intake noise and resonance effects.

Low pressure drop can be ensured by suitable pipe sizing restricting intake flow velocity to:-

(i) 5–6 metres/sec (16–20 ft/sec) in the case of single-acting compressors

(ii) 6–7 metres/sec (20–22 ft/sec) in the case of double-acting compressors

Combined with a venturi-type intake silencer these intake velocities can be increased by up to 10%, this form of treatment becoming increasingly used. A typical arrangement of intake piping with venturi silencer is shown in Fig 1.

Due consideration must be given to ventilating the compressor house to dissipate heat generated by the compressor and driver. This is particularly necessary in the case of air-cooled units. With water cooled compressors, or air-cooled compressors with water-cooled intercooler, the amount of radiant heat dissipated is usually negligible since the majority of heat is carried away by the cooling water.

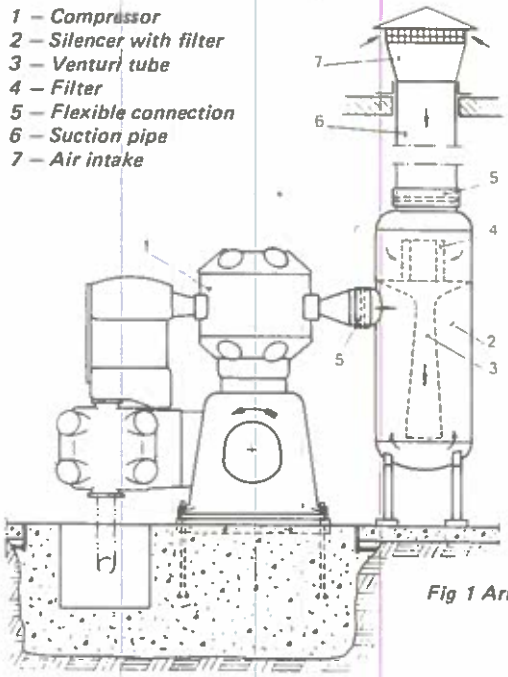


Fig 1 Arrangement of air intake pipe and venturi silencer. (Atlas Copco)

Energy Recovery

With water cooled compressors it is possible to recover a high proportion of the intake energy in the form of hot water which can be used for heating, feed water for boilers, *etc* in a large installation. A typical system for heat recovery is shown in Fig 2. With air cooled compressors the hot air leaving the intercooler and aftercooler can be diverted directly to an adjacent room for space heating (Fig 3).

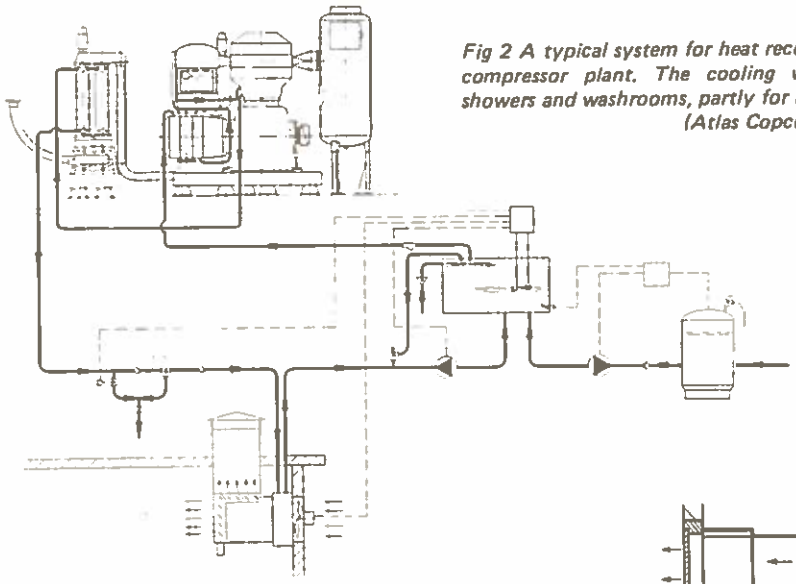


Fig 2 A typical system for heat recovery from a water-cooled compressor plant. The cooling water is used partly for showers and washrooms, partly for heating adjacent premises. (Atlas Copco).

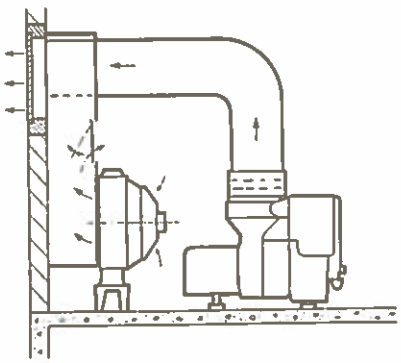


Fig 3 Heat recovery from an air-cooled compressor installation.

Foundations

Installation is seldom a problem with compressors below a power rating of 25–30 kilowatts. Larger machines, however, may need special foundations, the size and cost of such foundations increasing with compressor size, particularly in the case of reciprocating compressors with large out-of-balance forces. Very considerable savings are possible in these cases with 'L', 'V' or 'W' configurations due to the considerable reduction of such forces with these layouts. Large centrifugal and axial flow type compressors are far less troublesome since they are essentially balanced machines and the foundations are required to take little more than the weight of the machine and driver.

Larger compressors will, however, require the use of lifting or hoisting equipment to erect them, and also to dismantle or remove sections for inspection and maintenance.

Delivery Side

In the case of medium and large size compressors delivery pipes can be laid on the floor, or preferably in trenches covered by floor plates. In both cases pipes should be properly supported with sufficient freedom for thermal expansion. Long pipes can also be suspended flexibly from wall brackets. They should never be attached solidly to walls or ceilings.

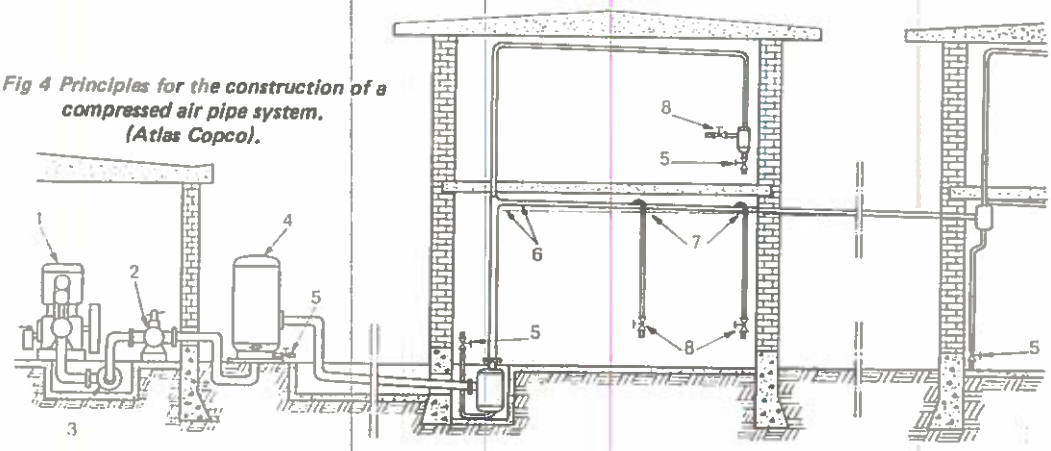


Fig 4 Principles for the construction of a compressed air pipe system. (Atlas Copco).

1 — Compressor
2 — Aftercooler
3 — Discharge pulsation damper
4 — Air receiver
5 — Condensate draining valve
6 — Inclination towards condensate drainer
7 — Branch pipe connections
8 — Outlet valve

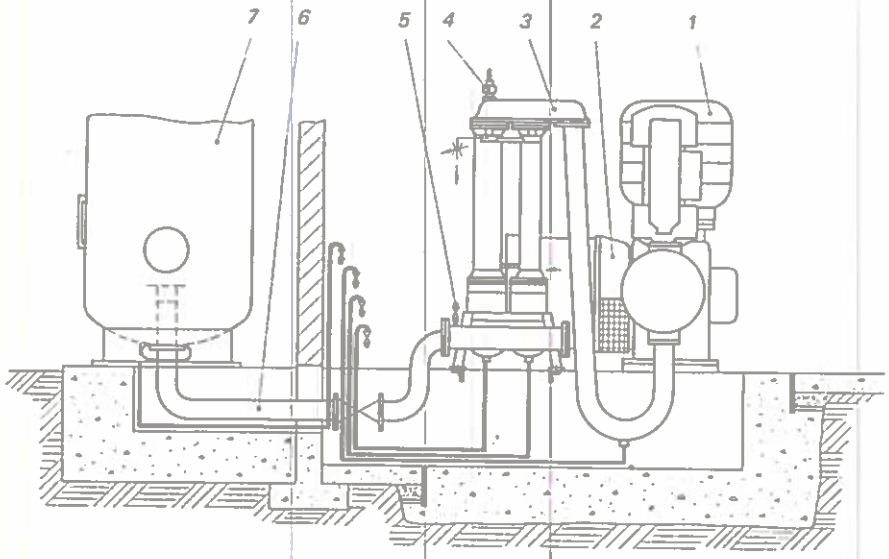


Fig 5 Compressor with aftercooler and air receiver. (Atlas Copco).

1 — Compressor
2 — Motor
3 — Aftercooler
4 — Safety valve
5 — Bleed-off valve
6 — Delivery pipe
7 — Air receiver

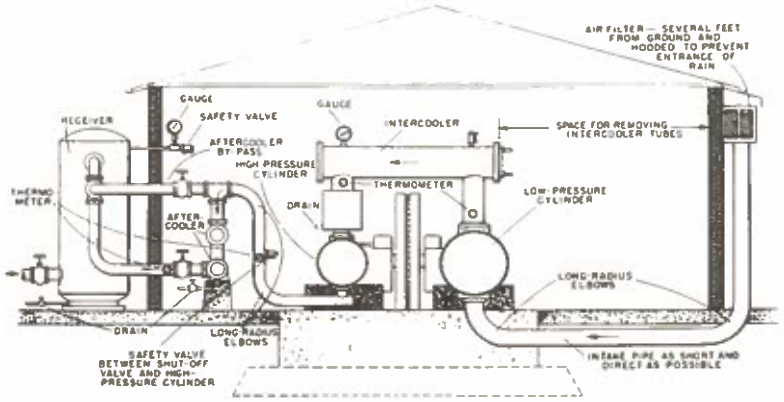
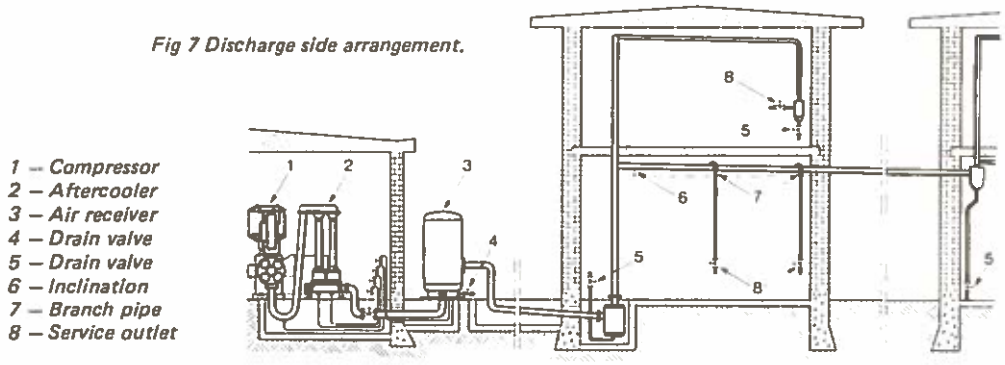


Fig 6 Typical two-stage compressor installation.

Fig 7 Discharge side arrangement.



Where a tuned delivery pipe is specified, its dimensions are specified by the compressor manufacturer and must be strictly adhered to. If this is not practicable, then a pulsation damper must be installed with the shortest possible delivery pipe between the damper and aftercooler.

Ideally delivery pipes for medium-sized compressors and large stationary compressors should be welded and provided with flanged connections for fittings. With smaller compressors threaded pipe connections are more commonly used.

A complete compressor installation normally involves compressor and drive motor, aftercooler, moisture separator and receiver. The latter needs to be located in a cool area and so is frequently positioned outside the compressor chamber, as in the typical arrangements shown in Figs 4 and 5. However, an important point to be considered is that the length of the discharge pipe between compressor and receiver should be kept as short as possible.

Where an isolating valve is fitted in the delivery pipework the line on the compressor side should be protected by a safety valve of sufficient capacity to pass the full output of the compressor without the pressure rising more than 10% above the set blow-off pressure.

Two further installations, typical of good practice, are shown in Figs 6 and 7. See also chapters on *Compressed Air Treatment* and *Compressor Controls.*

Air Receivers

A COMPRESSOR plant is normally followed by one (or more) pressure vessels known as air receivers which provides the following functions:

(i) Equalization of pressure variations in the supply lines.

(ii) Storage of compressed air to meet heavy demands in excess of the compressor capacity.

(iii) A source of additional cooling and a collection point for residual condensate and oil droplets.

(iv) Prevention of rapid loading and unloading of the compressor in short cycle duties.

Basic requirements of an air receiver are that it should be of adequate capacity to match the demands of the system; and must meet pressure vessel regulations applicable (*eg* classification according to Lloyd's).

Storage Capacity

The useful storage capacity of a receiver is the volume of free air which can be drawn from the receiver over the required, or permissible, working pressure range. The size of the receiver must, therefore, be proportioned to provide the storage capacity required. For constant demand the volume of the receiver need not be greater than the output of the compressor, in terms of output at working pressure, since the receiver need only operate as a pressure damper to maintain uniform pressure. Where the consumption is only intermittent, or variable, the receiver must have a margin of storage capacity relative to the compressor output, to adjust to demand and damp out pressure fluctuations.

For constant demand the theoretical size of receiver required is:-

$$\text{Volume of receiver } (V_R) = \frac{C_o \times \text{atmospheric pressure}}{P_a}$$

$$= \frac{C_o \times \text{atmospheric pressure}}{Pg + \text{atmospheric pressure}}$$

where P_a = working pressure, absolute

= gauge pressure Pg + atmospheric pressure

TABLE I – TYPICAL RECEIVER SIZES

Receiver Volume		Matching Compressor Capacity (FAD)			
		Constant Speed Control*		Stop-Start Control†	
ft^3	m^3	ft^3/min	m^3/min	ft^3/min	m^3/min
4.5	0.125	45	1.25	25	0.71
11	0.31	110	3.1	55	1.6
20	0.57	190	5.4	100	2.8
34	0.96	340	9.6	170	4.8
55	1.6	550	15.6	275	7.8
100	2.8	1 000	28	500	14
150	4.2	2 000	57	1 000	28
225	6.4	3 000	85	1 500	42.5
325	9.2	4 500	127	2 250	63
430	12.2	6 000	170	3 000	85

Note:

* This is consistent with the general rule that a receiver capacity of one-tenth the compressor FAD should be adequate for average systems.

† This is consistent with the general rule that with automatic stop-start control the receiver size can be one half that necessary with constant speed control.

Note also that with increasing compressor capacity the relative size of receiver can usually be reduced from the above proportions.

If the demand is constant and steady, the size need not be proportioned to the output of the compressor, since under such conditions the function of the receiver is merely to damp out pulsation pressures, so as to maintain a constant pressure. Normally, however, a receiver volume matched to the compressor output is regarded as the minimum size required (calculated on the basis of the compressor output in one minute). Alternatively, for working pressures up to 9 bar and normal air consumption conditions, a receiver volume of about six times the compressor capacity in the same volume units per second may be adopted, *ie*

$$V_R \text{ (litres)} = 6 \times \text{compressor capacity, litres/sec}$$

For pressures above 9 bar, higher pressure drops can usually be tolerated and the receiver sizes can be reduced.

Another general formula which may be used, based on demand and acceptable pressure drop is

$$V_R \text{ (litres)} = \frac{\text{demand per operation (litres)}}{\text{pressure drop (bar)}}$$

In this case it is necessary that the compressor be large enough to recharge the air receiver up to the original pressure in the intervals between demand.

Variable Demand

If the demand is variable, then a larger receiver volume is necessary to prevent rapid fluctuations of pressure. The assessment of a suitable size is largely arbitrary for general applications, and is commonly taken as three times the minimum size (*ie* three times V_R calculated on the basis of compressor output).

TABLE IIA – USEFUL CAPACITY OF AIR RECEIVERS (cubic metres of free air)

Receiver Volume m^3	Permissible Pressure Drop – bar							
	0.25	0.5	0.75	1	1.25	1.5	2	2.5
1	0.25	0.5	0.75	1	1.25	1.5	2	2.5
2	0.5	1	1.5	2	2.5	3	4	5
3	0.75	1.5	2.25	3	3.75	4.5	6	7.5
4	1	2	3	4	5	6	8	10
5	1.25	2.5	3.75	5	6.25	7.5	10	12.5
6	1.5	3	4.5	6	7.5	9	12	15
7	1.75	3.5	5.25	7	8.75	10.5	14	17.5
8	2	4	6	8	10	12	16	20
9	2.25	4.5	6.75	9	11.25	13.5	18	22.5
10	2.5	5	7.5	10	12.5	15	20	25

TABLE IIB – USEFUL CAPACITY OF AIR RECEIVERS (cubic feet of free air)

Receiver Volume ft^3	Permissible Pressure Drop – lb/in^2							
	5	10	15	20	25	30	35	40
1	0.34	0.7	1	1.3	1.7	2	2.3	2.7
2	0.68	1.4	2	2.7	3.3	4	4.7	5.4
3	1.0	2.0	3	4	5	6	7	8
4	1.4	2.8	4	5.2	6.6	8	9.4	10.8
5	1.7	3.4	5	6.7	8.4	10	11.7	13.4
6	2.0	4.0	6	8	10	12	14	16
7	2.4	4.8	7	9.4	12	14	16	19
8	2.7	5.4	8	11	13	16	19	21
9	3.0	6.0	9	12	15	18	21	24
10	3.4	6.8	10.2	13.5	17	20	23.5	27
20	6.8	13.6	20.4	27	34	40	47	54
30	10.2	20.4	30.6	40.5	51	61	71	81
40	13.6	27.2	40.8	54	68	82	94	108
50	17.0	34.0	51	68	85	102	118	136
60	20.4	40.8	61.2	81.5	102	122	143	163
70	23.8	47.6	71.4	95	119	143	167	190
80	27.2	54.4	81.6	109	136	163	190	217
90	30.6	61.2	91.8	122	153	183	216	244
100	34.0	68.0	102	136	170	204	238	272

Storage capacity and the time over which this capacity can be delivered are given by the relationship

$$V_R = \frac{P_o \times V_d \times t}{\Delta P}$$

or

$$t = \frac{V_R \, \Delta P}{P_o \times V_d}$$

where

V_d = air volume requirement (volume of air drawn)
t = time in minutes over which this air is to be drawn

This formula is less likely to be misinterpreted if expressed in the form

$$t = \frac{V_R \, (P_1 - P_2)}{P_o \times V_d}$$

where

P_1 = initial receiver pressure
P_2 = minimum permissible working pressure

If a pressure drop of 1 bar is acceptable, this formula simplifies to

$$t = \frac{V_R}{V_d}$$

These formulas assume that no air is supplied to the receiver during the time interval in which air is being drawn off. If the receiver is being continuously supplied with air from the compressor at a constant rate during this interval, the appropriate formula is

$$\text{time (minutes)} = \frac{V_R \times \Delta P}{V_a \times (V_d - S)}$$

where S is the delivery of the compressor, in the same units as V_d

$$t(\text{seconds}) = \frac{0.07 V_R \times \Delta P}{V_d}$$

where

V_R = receiver volume in metres^3
V_d = delivery required in metres^3
ΔP = pressure drop in bar

Thus, for a permissible pressure drop of 0.7 bar

$$t = \frac{0.05V_R}{V_d} \quad \text{(approx)}$$

For a permissible pressure drop of 1 bar

$$t = \frac{0.07V_R}{V_d}$$

In practice it may be necessary to restrict the receiver inlet so that an excessive amount of air may not be withdrawn under heavy demand. Thus, even with continuous charge, the charging rate may be relatively low in such cases, *ie* when operating under heavy demand, and it may be safer to size the receiver on the basis that it does not receive any charge during the time of heavy demand.

For a given compressor and a given receiver the maximum number of starts, with the compressor operating an automatic stop-start control, will be obtained when the demand (V_d) is half the FAD of the compressor. Basically, the compressor running time will be thirty minutes in every hour, and during the charging period half of the FAD will be absorbed by the demand and the other half will go to charge the receiver. The actual number of stops and starts will depend on the size of the receiver.

$$\text{Free air available from receiver} = \frac{V_R \Delta P}{P_a}$$

$$V_d = S/2$$

where S = FAD of compressor

Time taken to use up receiver air

$$= \frac{V_R \Delta P}{P_a} \times \frac{1}{S/2}$$

$$= \frac{2V_R \Delta P}{P_a \cdot S}$$

This represents one half the total time

$$\text{Hence total time} = \frac{4 V_R \Delta P}{P_a \cdot S} = \frac{60}{n}$$

$$\text{where } V_R = \frac{15P_a S}{n \Delta P}$$

where n = number of stops and starts

For a pressure drop of 1 bar (14.5 lb/in^2) $V_R = \frac{15S}{n}$

For a pressure drop of 0.7 bar (10 lb/in^2) $V_R = \frac{22S}{n}$

Air Receiver Design

The usual form of pressure vessel is a cylindrical shell with dished ends. Material stress in the cylinder walls can be determined as

$$S = \frac{PD}{2t}$$

or

$$t = \frac{PD}{R}$$

where

S = material stress
P = internal pressure
D = overall diameter of shell
R = radius of shell
t = wall thickness

This formula is valid where P is the maximum permissible internal pressure consistent with a maximum permissible material stress, provided:-

(i) The material stress figure allowed is below the limit of proportionality of the material.

(ii) The material stress figure is determined for the maximum service temperature of the pressure vessel.

The formula does not make any allowance for stress raisers, effects of corrosion, and method of construction. These can be allowed for separately, or taken into account in determining the maximum permissible material stress.

Recommended practice is to base maximum permissible material stress on the upper yield point of the material, with a factor of safety applied to this value, viz

$$\text{Maximum permissible stress} = \frac{\text{upper yield point}}{f}$$

For ISO/TC semi-killed and killed structural steels, f = 1.92; *ie* maximum permissible material stress = 0.521 x upper yield point. For pressure vessel steels and ISO/TC17 structural steels, f = 1.6; *ie* maximum permissible material stress = 0.625 x upper yield point.

Typical upper yield point values are shown in Tables IIIA, IIIB and IIIC.

With *welded* construction an additional safety factor is normally called for — typically 0.8 for conventional manual and machine welding with normal inspection. Where the welding is subject to strict control and inspection, a higher value of 0.9 may be used.

An additional allowance may also be made for corrosion reducing the effective thickness with age. This takes the form of a simple addendum to calculated minimum thickness, a typical value being 0.75 mm (0.03 in).

TABLE IIIA – PRESSURE VESSEL STEELS

Specification	Upper Yield Point at Temperature of 20°C		100°C		150°C		200°C	
	lb/in²	bar	lb/in²	bar	lb/in²	bar	lb/in²	bar
UK								
BS 1501 –								
151 Grade 23B	29 300	2 060	25 000	1 760	24 500	1 720	23 000	1 640
161 Grade 23B	29 300	2 060	25 000	1 760	24 500	1 720	23 300	1 640
211 Grade 26B	34 300	2 410	29 700	2 090	29 300	2 060	28 700	2 020
211 Grade 28B	37 000	2 600	32 000	2 250	31 600	2 220	30 800	2 170
Germany								
DIN 17155 –								
HI	32 700	2 300	–	–	–	–	25 690	1 800
HII	37 000	2 600	–	–	–	–	29 900	2 100
HIII	39 800	2 800	–	–	–	–	34 100	2 400
17Mn4	41 250	2 900	–	–	–	–	35 500	2 500
Sweden								
INSTA –								
1B, C, D (SIS 1330)	34 100	2 400	32 000	2 250	30 600	2 150	29 900	2 000
2B, C, D (SIS 1330)	37 000	2 600	34 800	2 450	33 400	2 350	32 000	2 200
3B, C, D (SIS 1430)	39 800	2 800	37 700	2 650	35 600	2 550	34 800	2 400
4B, C, D (SIS 2101)	48 400	3 400	46 200	3 250	44 800	3 150	44 100	3 000
USA								
ASTM A 285B	27 000	1 900	25 400	1 800	24 200	1 700	20 000	1 400
ASTM A 201A	30 000	2 100	27 700	1 950	27 000	1 900	25 000	1 750

TABLE IIIB – ISO/TC 17 SEMI-KILLED AND KILLED STEELS

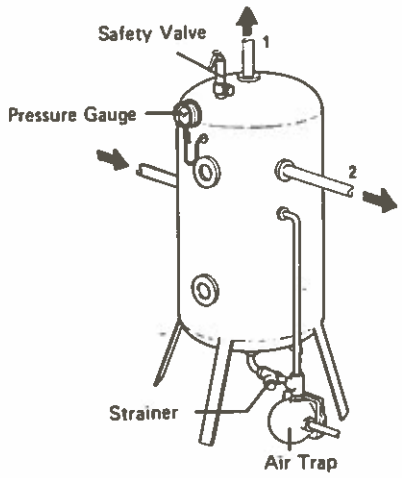
Designation	Upper Yield Point at Temperature of 20°C		125°C		200°C	
	lb/in²	bar	lb/in²	bar	lb/in²	bar
Fe 37B	34 000	2 400	29 500	2 070	–	–
Fe 42B	37 000	2 600	33 000	2 320	–	–
Fe 44B	40 000	2 800	36 000	2 520	–	–

TABLE IIIC – ISO/TC 17 STRUCTURAL STEELS

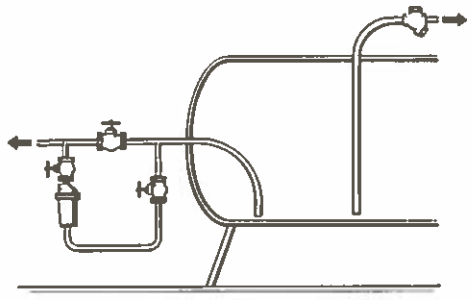
Designation	Upper Yield Point at Temperature of 20°C		125°C		200°C	
	lb/in²	bar	lb/in²	bar	lb/in²	bar
Fe 37C–D	34 000	2 400	–	–	25 600	1 800
Fe 42C–D	37 000	2 600	–	–	30 000	2 100
Fe 44C–D	40 000	2 800	–	–	33 000	2 300
Fe 50D	51 000	3 600	–	–	42 500	3 000

Shapes and Volumes

The usual form of receiver shape is cylindrical with dished ends – see Fig 1. The type with one end concave can be placed directly on a plinth on a floor. The type with convex ends can be fitted with feet for vertical or horizontal mounting.



Typical vertical receiver with air trap.



Small horizontal receiver with alternative trapping arrangements.

Fig 1
(Sarco)

TABLE IV – CAPACITY OF AIR RECEIVER

Shell inner diameter mm	Shell capacity per 300 mm length litres	Dished end capacity litres
150	5.4	0.85
200	9.9	1.98
230	15.0	3.12
250	26.6	4.52
380	34.8	5.66
450	50.0	6.51
610	74.5	10.5
760	139	17.6
914	201	31.0
1066	272	49.5
1220	356	87.5
1372	417	141
1525	510	226
1675	670	300
1830	820	396
1980	940	480
2134	1090	595
2286	1250	690
2438	1420	792

The volume of such shapes can readily be determined by calculating the volume of an equivalent flat-ended cylinder in the case of Type 1 with convex/concave dished ends; or in the case of convex/convex ends, adding twice the dished end capacity to this cylindrical volume. Table IV provides direct solutions for shell diameters from 150 mm (6 inches) to 2438 mm (96 inches).

End Shape Thickness

The stress formula in the case of dished ends is modified by a shape factor (F), when the solution for shell thickness becomes:

$$\text{Minimum thickness of shell (t)} = F\,\frac{RP}{S} + e$$

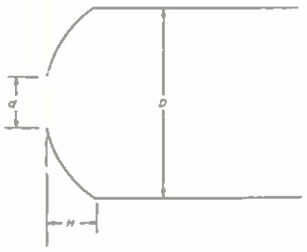


Fig 2

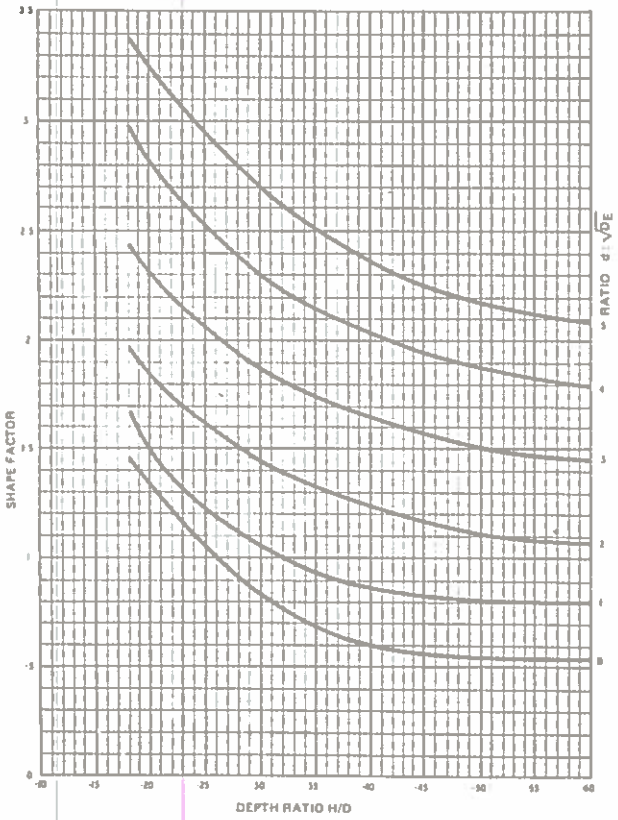


Fig 3

The shape factor is related to the depth ratio of the dished (drumhead) end, expressed as the height of the drumhead (H) divided by the diameter (D) — see Fig 2. It also takes into account the weakening effects of any opening which may be necessary, the opening being of diameter d. In the absence of an opening, the shape factor is substantially less than unity,with increasing depth ratio. If there is an opening the effect is most readily expressed in terms of the ratio.

$$d : \sqrt{D} \quad \text{(as in Fig 3)}$$

Solutions in such terms are based on the opening being unreinforced.

It is necessary to establish a maximum permissible diameter for unreinforced openings because the shape factor increases rapidly with increasing diameter of opening for a given shell diameter. This is specified in the PNEUROP code in terms of a factor K related to the product of drumhead diameter and thickness. The value of K is calculated from the formula

$$K = \frac{DP}{1.82St}$$

where S is the maximum permissible material stress.

Maximum permissible diameters for unreinforced openings are then given by Fig 4.

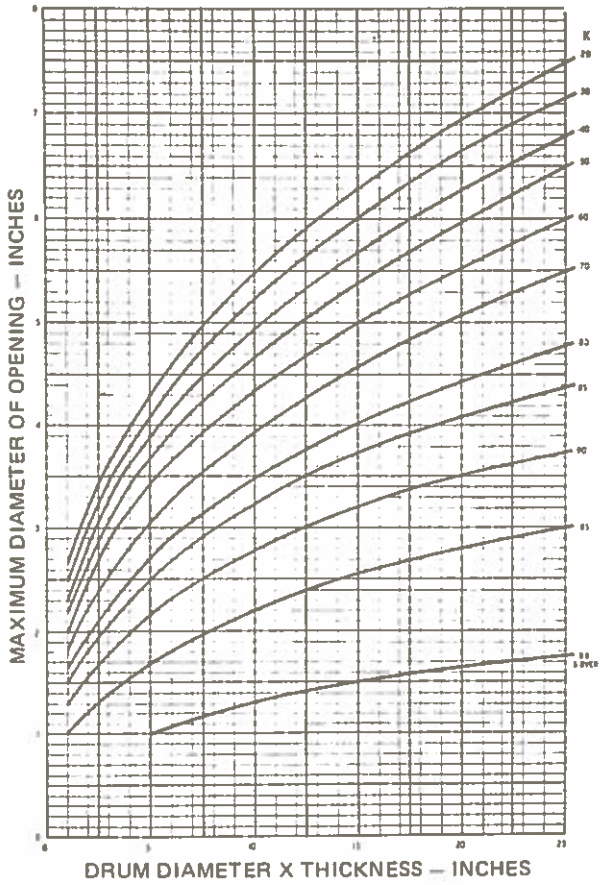
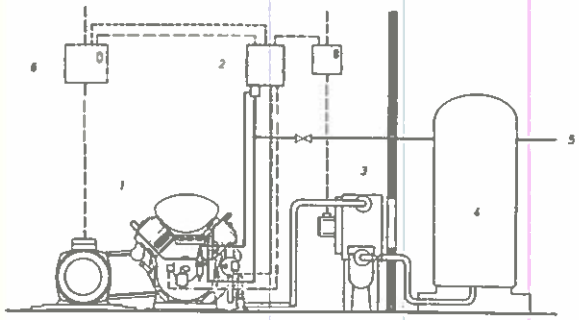


Fig 4

Air Receiver Location (see Fig 5)

The receiver is normally placed as close to the compressor as possible in order to keep the discharge pipe short and eliminate surge effects. In practice the receiver follows the intercooler and separator and may, in fact, be located in a separate block in order to be sited in a cool location. This does not nullify the requirement above.



1 – Compressor/motor unit
2 – RD regulator
3 – Aftercooler
4 – Receiver
5 – Delivery line
6 – Starting equipment

Fig 5 Compressor installation showing recommended position of receiver. (Atlas Copco)

A stop valve should not be fitted between the compressor and receiver unless a safety valve is also incorporated between this valve and the compressor, to guard against the possibility of the compressor being started up with the stop valve closed. When a compressor is connected to an air mains in common with other compressors, a safety valve must be fitted between the compressor and the first stop valve, for similar reasons.

Heat Exchangers and Aftercoolers

ALL PRACTICAL compressors work under adiabatic conditions with some heat loss. The amount of air heating is, however, considerable. This may not be an immediate disadvantage at low compression ratios, except that the air will be saturated with water and aftercooling could remove much of this. Unless the machine itself specifically demands cooling, therefore, a low compression ratio compressor may work satisfactorily uncooled (*eg* a single-stage centrifugal blower). If the machine does demand cooling (*eg* reciprocating compressors), the necessary heat exchange may be provided by finned cylinders with or without forced circulation. The chief limitation which applies then is that if the machine is not adequately cooled it will transfer heat to the inducted charge which can seriously reduce the efficiency of the compressor.

In general cooling may be advantageous, or necessary, to achieve the following:

(i) Maintain working clearances on machine components.

(ii) Limit the amount of heating applied to the inducted charge, thus improving compression efficiency.

(iii) Cool the compressed air between stages to improve compression efficiency.

(iv) Cool the delivered air to remove water (aftercooling).

Interstage Cooling

Where more than one stage of compression is involved, it is desirable from the point of view of maximum efficiency that each stage does an equal amount of work. This can only be realized if the temperature of the air fed into the second stage is similar to that of the air inducted into the first stage. Machine cooling will remove only a proportion of the heat generated by compression, because only a proportion of the air passing through the compressor will come into contact with the cooled surfaces of the cylinder walls. Thus, with an efficient water cooling system applied to a reciprocating compressor cylinder, only about 10–15% of the heat generated by compression will actually be removed. Cooling of the air itself, or intercooling, is thus necessary between stages, to bring the air temperature down to more acceptable levels. This represents an independent problem, involving the use of a separate heat exchanger.

Aftercoolers

An aftercooler is a heat exchanger used to cool compressed air after leaving the compressor. Cooling reduces the air temperature, and precipitates out water droplets which can be drained off

(all aftercoolers incorporate a moisture separator for this purpose). Aftercooling reduces the volume of air available, but this represents no real power loss.

As a general rule, air-cooled aftercoolers are used with air-cooled compression; and water-cooled aftercoolers with water-cooled compressors. Temperature of the discharged compressor air is normally about 12—16 K above the ambient air temperature when air-cooling is used; and 10 K above the water temperature when water-cooling is used.

Air Cooling

In the case of very small single-stage compressors, natural air cooling may be sufficient, using finned cylinders. The performance of such a compressor can be expected to vary somewhat with changes in ambient temperature, as the cooling available is only relative and not specific. Forced draught cooling is generally preferred, *eg* generated by an engine driven fan or from a power take-off on the compressor shaft, as this can provide both more effective heat dissipation and more consistent heat transfer performance. If the compressor is located in a warm atmosphere it may be advantageous to draw the cooling air from a lower temperature source, *eg* through ducting led to the outside of the building. The degree of cooling required by a compressor is appreciably higher than that considered the optimum with an internal combustion engine. The aim is to lower the compressor temperature as far as possible, rather than to maintain it at a steady, but fairly high 'running' temperature.

Forced draught air cooling may be adequate for both small and large compressors, particularly if the latter are low capacity or multi-stage machines. With multi-stage machines, the increase in temperature per stage is lower than that of a single-stage machine. Air cooling may, however, have distinct limitations where the compressor has a moderate to high capacity and is run continuously.

Water Cooling

Most larger single-stage compressors, and the bulk of multi-stage compressors, are water cooled. In the case of reciprocating compressors, the principle adopted is basically similar to that of internal combustion engine cooling, using water jackets for the cylinder barrels and cylinder heads. These can either be cored into the cylinder and head castings, or added as fabricated jackets. An alternative method sometimes used is to immerse the whole cylinder in a bath.

The temperature of the cooling water should be as low as possible and the rate of water change high to reduce the compressor temperature as much as possible. In a closed system this will demand the use of a further heat exchanger to cool the circulating water and maintain lower water inlet temperature, which again can follow automobile practice — *eg* a radiator exposed to a forced draught provided by a fan. In larger installations other types of water cooler, such as cooling towers, may be used to ensure a supply of cold water to the cooler inlet. Typical water cooling requirements are shown in Tables IA and IB.

Centrifugal compressors are generally cooled by one of three methods:

(a) *Diaphragm cooling* — which implies that the return and diffuser passages are water jacketed with a supply of coolant water through these jackets.

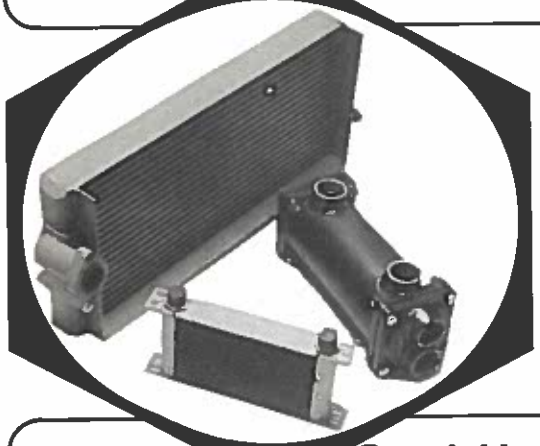
(b) *Injection cooling* — which is direct cooling produced by spraying cold water into the air passing through the machine.

(c) *Intercooling* — applied between stages.

Diaphragm cooling has the advantage that the cooling of each stage can be separate and regulated or adjusted accordingly, to give close control over temperature.

Injection cooling has the advantage that it reduces the temperature rise of the gas during compression without removing heat from the system.

HEAT EXCHANGERS
FROM SERCK

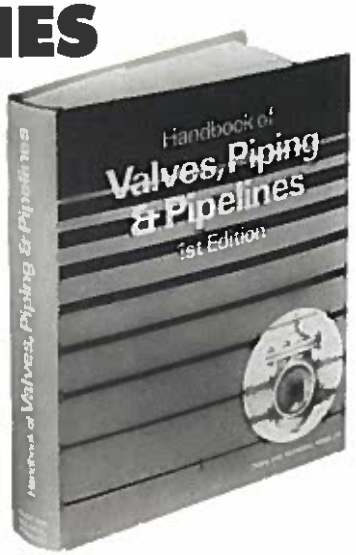


Team up
with the Serck range of
air and water cooled heat
exchangers.
If you want a tough, dependable
unit suitable for high pressure work,
look no further.
If you want an efficient
lightweight aluminium unit,
Serck have the answer.
Whatever your application, whatever
your market, team up with Serck
and get the specialists
on your side.

Serck Heat Transfer
Warwick Road, Birmingham B11 2QY England.
Telephone: 021-772 4353 Telex: 337341, 339805

HANDBOOK OF VALVES, PIPING & PIPELINES
1ST EDITION

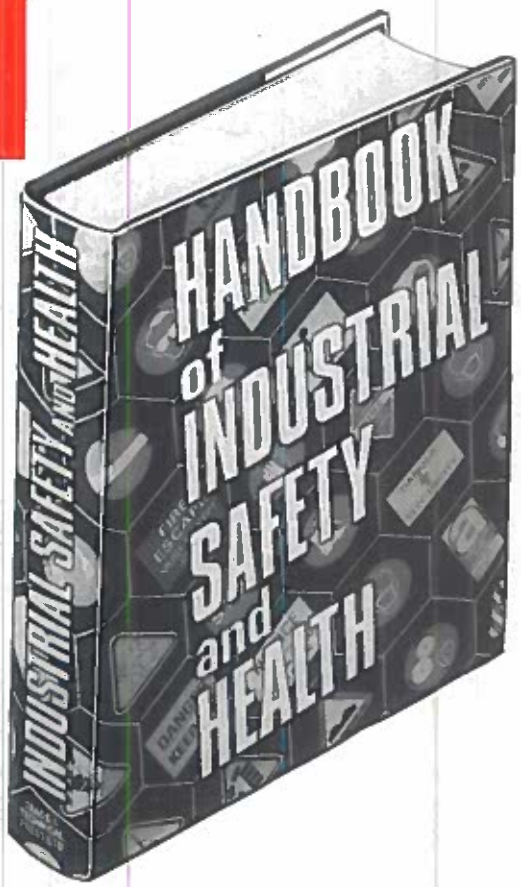
OVER 500 PAGES, HAND BOUND, HARD CASE COVER, GOLD BLOCKED, 2,000 DIAGRAMS, TABLES, CHARTS, ILLUSTRATIONS.



Contents include: Sect 1 — Introductory:- Piping Services, General Summary and Classification of Valves, Valve Nomenclature. Sect 2 — Types of Valves. Sect 3 — Valves — Design and Construction. Sect 4 — Valve Actuators. Sect 5 — Performance, Fluid Transport, Frictional Losses, Pressure Drop. Sect 6 — Pipelines. Sect 7 — Services (Valves). Sect 8 — Services (Pipelines). Sect 9 — Materials. Sect 10 — Guides and Standards. Sect 11 — Data. Sect 12 — Buyers Guide.

TRADE AND TECHNICAL PRESS LTD., CROWN HOUSE, MORDEN, SURREY, SM4 5EW, ENGLAND.

NEW



Industrial injury and disease is, without doubt, one of the biggest causes of lost working days. If these lost days are to be reduced and industry become more efficient and productive, then a greater understanding of health and safety at work is required. Based on their wealth of experience in the technical publishing field, the Trade and Technical Press Ltd., have published the 1st Edition HANDBOOK OF INDUSTRIAL SAFETY AND HEALTH. More than 650 pages, 1500 illustrations, diagrams and charts, this comprehensive reference work provides essential information vitally important to Top Management, Chief Engineers, Works Managers, Safety Officers, etc. Contents include:-

SECTION 1 — THE CONTROL AND MANAGEMENT OF SAFETY AND HEALTH
Legal Aspects — Standards and the EEC — HASAWA and Enforcement — Product Liability — Safety Policies — Safety and Health Management Techniques — Risk Management — Multi occupancy — Total Loss Control — Compensation Claims, etc, etc.

SECTION 2 — RESPONSIBILITIES AND DUTIES
General duties of Employer — Safety Officials — Representatives and Committees — Authorities and Organisations — Roles of Specialists — Employees' responsibilities — Accident Investigation, etc, etc.

SECTION 3 — THE SAFE PLACE
Design Safety — Safety Inspections — Site Safety — Check list — Electrical Safety — Lighting — Flooring and surfaces — British and European Standard signs and labels — Notices, etc. etc.

SECTION 4 — SAFETY OF PLANT AND SYSTEMS OF WORK
Design of Systems — System safety and analytical techniques — Machine hazards and protection — Safety guards and barriers — Laser safety — Communications equipment and Visual Display Units — Pressure vessels — boilers, etc. etc.

SECTION 5 — OCCUPATIONAL SAFETY AND HEALTH
Safety Training and Instruction at all levels — Hygiene and Welfare requirements — Dust and Fume Monitoring and Control — Noise and Vibration Measurement and Control — Hearing Conservation — Asbestos precautions, handling and removal — Ionising radiation and detection — Gas detection and monitoring — Temperature and Pressure monitoring — Barrier Creams and Cleaners — Janitorial Equipment, etc, etc.

SECTION 6 — THE SAFE PERSON
Personal Protection — Protective Clothing — Eye Protection — Ear Protection — Headgear — Face and Body shields — Footwear — Gloves and Mitts — Safety Harnesses, etc. etc.

SECTION 7 — SAFE USE HANDLING STORAGE AND TRANSPORT
Kinetic Handling — Handling of Materials — Mechanical Handling — Lifting equipment — Ropes, hoists and tackle — Safety tools — The Hazchem Code — Warning devices and perception — Labelling — Securing Containment and transportation — Disposal of toxic waste — Information, advice, abuse and expected misuse, etc, etc.

SECTION 8 — FIRE AND EXPLOSION
General Fire Safety — Fire fighting equipment — Inflammable Substances — Explosion and implosion.

SECTION 9 — EMERGENCY EQUIPMENT
Emergency lighting and power — Breathing and Rescue Apparatus — Resuscitation Equipment — First Aid, etc, etc.

SECTION 10 — BUYERS' GUIDE
Products and Trade Names Index — Alphabetical list of Manufacturers with addresses, telephone numbers, telex and telegram addresses of Head Office, Works and Branches

TRADE & TECHNICAL PRESS LTD. CROWN HOUSE MORDEN SURREY SM4 5EW ENGLAND

TABLE IA – TYPICAL COOLING WATER REQUIREMENTS (Gallons per 100 cu ft free air).

Component	Pressure		Water Temperature			
	lb/in^2 (gauge)	bar(gauge)	16°C (60°F)	21°C (70°F)	(27°C (80°F)	32°C (90°F)
Compressors:						
Cylinder Jacket:						
Single-stage	40	2.8	0.5--0.6	0.7	0.8	0.9
	60	4.2	0.7--0.8	1.0	1.1	1.2
	80	5.6	0.9–1.2	1.3	1.4	1.5
	100	7.0	1.2–1.5	1.6	1.7	1.8
Two-stage	80–120	5.5–8.5	1.0	1.1	1.2	1.4
Intercoolers:						
In series with cylinder jackets	80–120	5.5–8.5	2.8	3.4	4.0	4.5
Separate	80–120	5.5–8.5	2.5	3.0	3.5	4.0
Aftercoolers:						
Single-stage	80–120	5.5–8.5	4.0	4.5	5.2	6.0
Two-stage	80–120	5.5–8.5	2.5	3.0	3.5	4.0

TABLE IB – TYPICAL COOLING WATER REQUIREMENTS (Litres per cubic metre free air)

Component	Pressure		Water Temperature			
	bar(gauge)	lb/in^2 gauge	16°C (60°F)	21°C (70°F)	27°C (80°F)	32°C (90°F)
Compressors:						
Cylinder Jacket:						
Single-stage	2.8	40	0.8–1.0	1.15	1.3	1.45
	4.2	60	12.–1.3	1.6	1.8	1.95
	5.6	80	1.45–1.95	2.1	2.25	2.4
	7.0	100	2.0–2.4	2.6	2.75	2.9
Two-stage	5.5–8.5	80–120	1.6	1.8	1.95	2.25
Intercoolers:						
In series with cylinder jacket	5.5–8.5	80–120	4.5	5.4	6.5	7.3
Separate	5.5–8.5	80–120	4.0	4.85	5.7	6.5
Aftercoolers:						
Single-stage	5.5–8.5	80–120	6.5	7.3	8.5	9.7
Two-stage	5.5–8.5	80–120	4.0	4.85	5.7	6.5

Intercooling has little advantage unless the stage compression exceeds about 3:1, when it can reduce power consumption. It can be applied between successive stages, or at intervals of two or three stages, as necessary.

Where the compressor itself is water cooled, then it is usual for the intercooler also to be water cooled. In this case the intercooler normally takes the form of a nest of straight tubes in a cylindrical shell. Cooling water is then fed through these tubes, passing several times up and down the length of the intercooler. Compressed air from the preceding stage flows through the shell and over the tubes, and is further directed across the bank of tubes by baffles as necessary .

Two-shell intercooled construction is often preferred as providing two stages of intercooling and higher heat transfer for the same amount of cooling water. Each shell is a complete heat exchange unit, with its own nest of tubes and supply of cooling water. The compressed air flow is through each unit in turn, *ie* series flow. For most efficient cooling, with the lowest water flow, the coldest water is supplied to the final unit.

Other types of cooler include co-axial pipe designs and submerged coils. In the latter case a continuous coil of tubing carrying the air is submerged in a tank through which cooling water is circulated.

Water cooled intercoolers may be used with air cooled compressors, although the more usual arrangements here would be an air cooled intercooler. Air cooled intercoolers may also be used with water cooled compressors.

An air cooled intercooler normally comprises one or more lengths of finned tubing between headers. These lengths may be straight or coiled, depending on the heat exchange area required. Cooling is then assisted by forced draught, *eg* generated by a fan.

All intercoolers will condense out water and so must be provided with a drain. Treatment varies from the fitting of a simple drain tap at the lowest point of the intercooler, or the fitting of a drain leg, to the incorporation of a specific separator or separator chamber in the design of the intercooler.

Heat Exchanger Calculations

In any heat exchanger the rate of heat exchange between the hot and cold media is given by

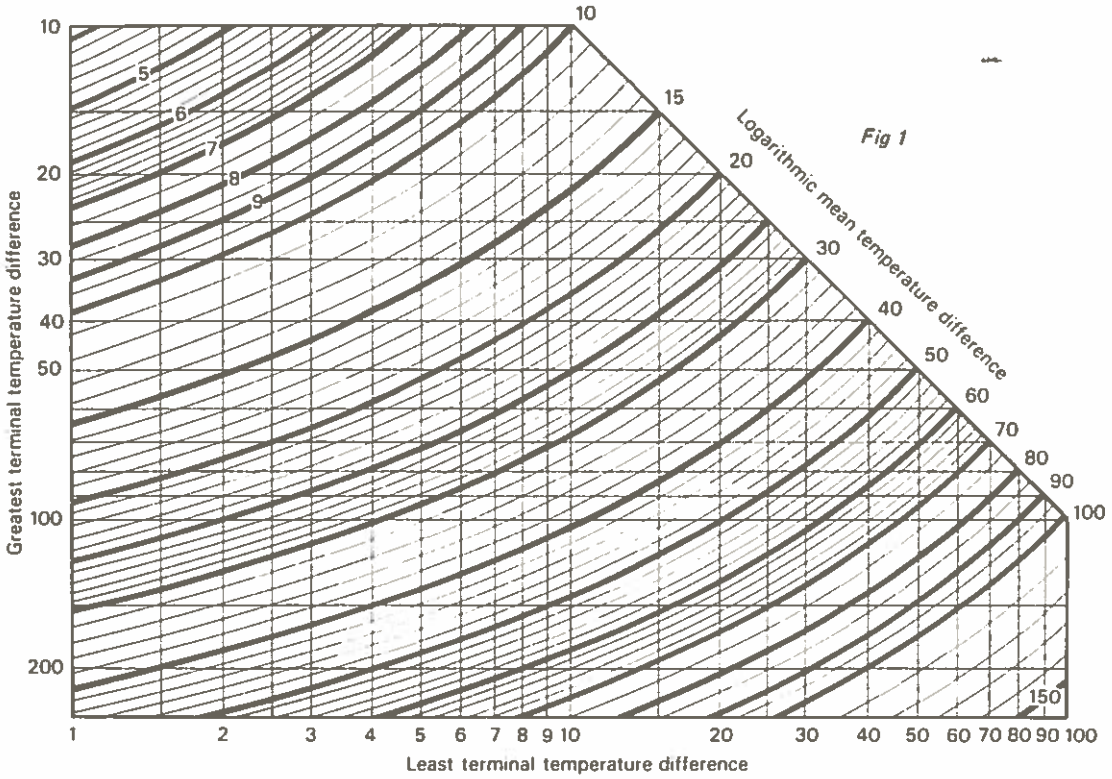
$$q/t = K.A.Tm$$

where q/t = rate of heat transfer in watts (W)
K = coefficient of heat transmission in $W/m^2.K$
A = effective cooling area in m^2
Tm = logarithmic mean temperature difference in K (see Fig 1)

An alternative simpler formula, particularly applicable to air cooling is

$$q/t \text{ (watts)} = A.R.K.V^{0.6}.S.\Delta t$$

where A = a constant (typically 0.0048)
R = exchange rate
K = average heat transfer coefficient in $kcal/hr/m^2/degC$
V = ventilating air speed in m/sec
S = total heat exchange area in m^2
Δt = temperature difference, in degC



The exchange rate (R) is a function of the ventilation present. In a fully enclosed system R = 1. In such a case, if the ventilating air speed (V) was zero (*ie* static air conditions inside the enclosure), there would be no heat transfer and the enclosure would simply assume the temperature of the fluid. In practice, the enclosure surface radiates some heat and V would assume a positive but small value, due to ascending air currents inside the enclosure, but the amount of cooling would be very small.

In a fully open or exposed system, surrounded by static air, the value of R approximates to 2. Any natural draught will increase this value until, with slightly forced ventilation, a value of 8 will be reached. With strong forced draught the value of R may rise to as high as 12 — *eg* this is typical of the exchange rate value achieved with a car-type radiator and fan cooling. A specific value of R can be calculated as $10 \times V^{0.6}$.

The average transfer coefficient is less readily determined as, besides the design of the heat exchanger and the materials used, this is also dependent, to some extent, on the fluid velocity and the temperature difference. Typical values normally lie between 3 and 8 kcal/hour/m^2/degC.

The ventilating air speed is not readily determined quantitively. A typical value is commonly assumed for static installations, based on empirical fan data.

The total heat exchange area can readily be manipulated. Since the power loss is directly proportional to the surface area S, it follows that heat exchange and cooling will be directly increased

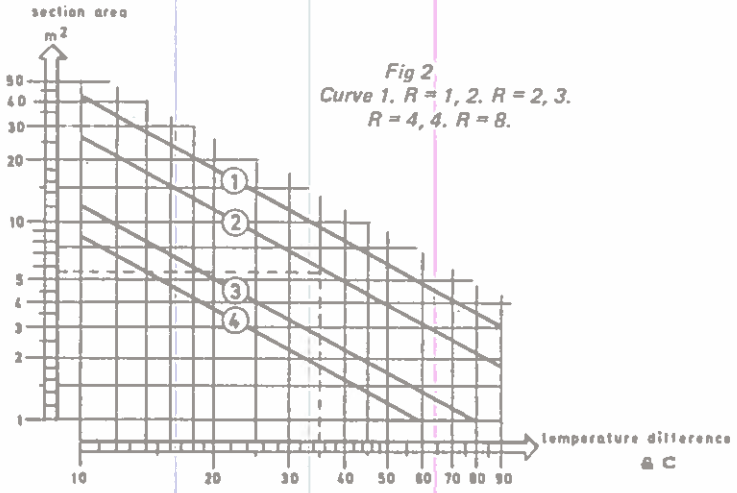


Fig 2
Curve 1. R = 1, 2. R = 2, 3. R = 4, 4. R = 8.

in proportion to the surface area available. For convenience, the heat exchange surface area required can be related to the temperature difference required and the exchange rate value in the form of a graph — Fig 2. This assumes typical or average values for the other parameters involved. Such a graph can be used directly to assess the area of the radiating surfaces required to produce the temperature difference desired, or to estimate the temperature difference likely to be achieved by a particular design.

Similar analysis can, of course, be applied to water cooling. In this case the same general equation applies, except that the average transfer coefficient is much higher. The ventilating speed then applies to the flow velocity of the circulating water. However, the heat gained by the water must itself be dissipated in a closed circuit, so that it can be returned to the inlet at the original (low) inlet temperature, if consistent cooling is to be maintained. Thus, although water cooling may dramatically reduce the size of heat exchanger required, *ie* total heat exchange area for direct cooling, a further heat exchanger is then required to cool the water.

Water cooling requirements are more generally solved on empirical or semi-empirical lines. The amount of heat exchange area available may be dictated by constructional considerations, or based on experience. Any final adjustment is then available through the flow rate. The resulting outlet temperature of the water must then be dealt with as necessary, *eg* by a second heat exchanger if the water is to be recirculated.

Compressor Acceptance Tests

BRITISH SPECIFICATIONS covering the testing of compressors to verify guaranteed performance were originally BS726 – Measurement of air flow for compressors and exhausters, and BS1571: 1949 – Acceptance tests for positive displacement compressors and exhausters. Both of these have been withdrawn and BS1571:1975 is now presented in revised metric form. Part 1 deals with type acceptance tests and Part 2 with simplified acceptance tests, calling only for measurements necessary to verify any guarantee given by the manufacturer to the purchaser; and in particular, capacity, shaft input power or specific energy consumption and speed. Following is an outline of the test procedure as given in BS1571:1975 Part 2.

Preliminary tests should be undertaken for the purposes of:

(a) determining whether the compressor and associated systems are in a suitable condition for an acceptance test to be conducted;

(b) checking of instruments;

(c) training of personnel.

After a preliminary test has been made, this test may, by agreement, be considered the acceptance test, provided that all requirements for an acceptance test have been met.

The governing mechanism must be maintained in its normal working position.

During the test, the lubricant, the adjustment of the lubricating pumps, lubricators or other lubricating means must comply with the operating instructions.

During the test no adjustments other than those required to maintain the test conditions and those required for normal operation as given in the instruction manual shall be made.

Before readings begin, the compressor shall be run long enough to assure that steady state conditions are reached so that no systematic changes occur in the instrument readings during the test.

The test report should include the following information:

General Information

(a) Type of machine; *eg* if reciprocating whether:

(1) vertical or horizontal,

(2) single- or double-acting,

(3) cylinders are air or water cooled,
(4) intercooled or aftercooled.
(5) single-stage, two-stage or multi-stage.

(b) Manufacturer's name and number.
(c) Date of manufacture.
(d) Type of prime mover.
(e) Name of manufacturer of prime mover and its number if applicable.
(f) Type of speed increasing or reducing gear (if fitted).
(g) Name of manufacturer of speed gear and its number.
(h) Type of coupling, if any, used for the test.

Guaranteed Performance on Test and/or Performance on Type Test

(a) Specified intake pressure and temperature.
(b) Capacity (FAD).
(c) Delivery pressure.
(d) Power consumption.
(e) Speed.
(f) Any other requirements of the standard, *eg* quantity of cooling water or temperature rise of cooling water.

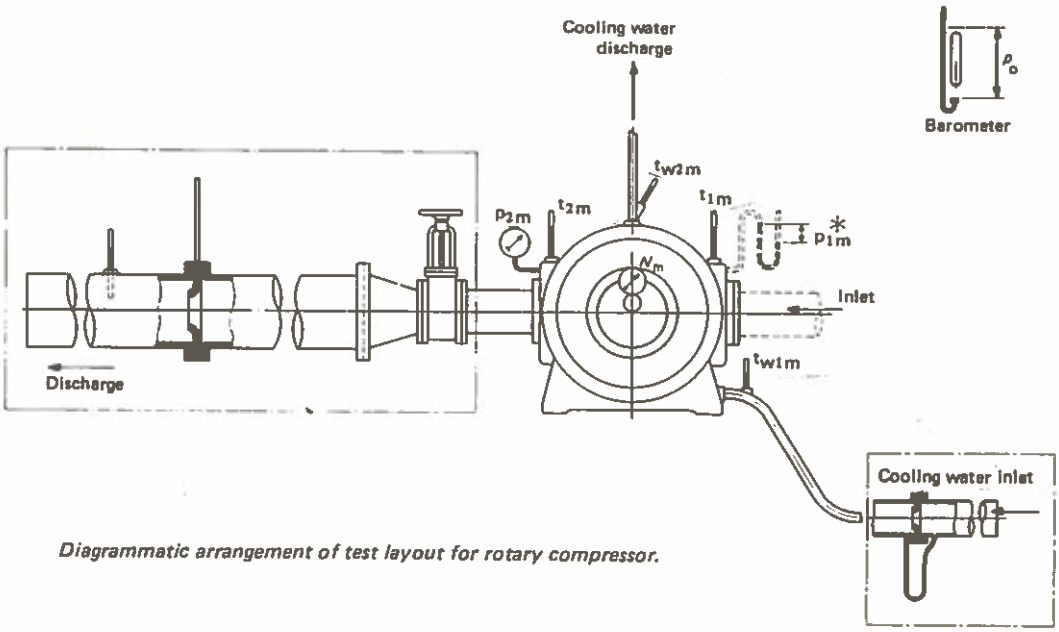
Test Observations

Only those measurements necessary to verify any guarantee given by the manufacturer to the purchaser.

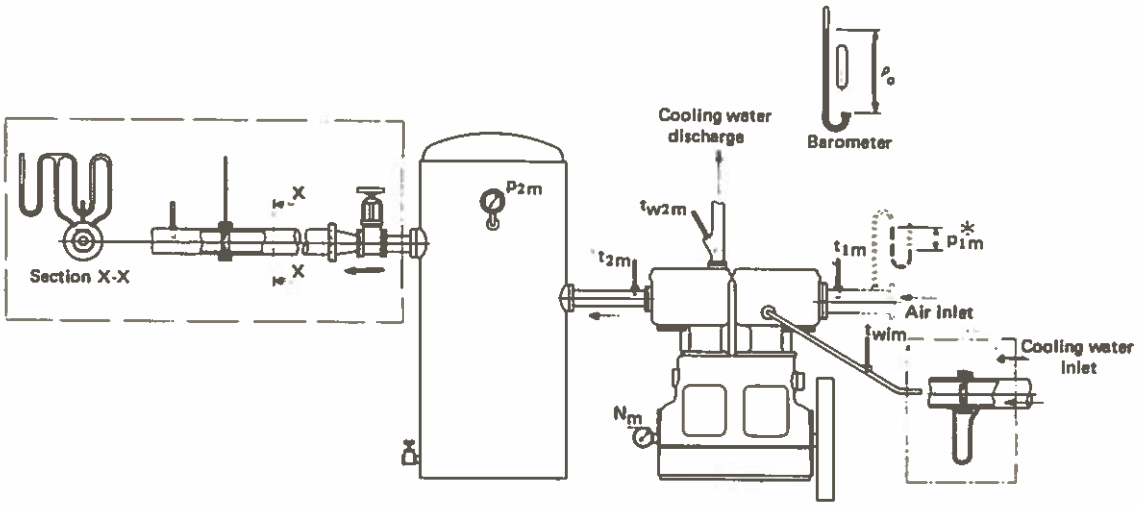
(a) Diameter of nozzle (mm).
(b) Speed (rev/min).
(c) Barometric pressure (mm Hg)
(d) Pressure at intake (absolute) (bar).
(e) Pressure drop across nozzle (mm H_2O).
(f) Pressure on downstream side of nozzle (mm Hg).
(g) Pressure at delivery (absolute) (bar).
(h) Temperature of air at intake (°C).
(j) Temperature of air at nozzle (°C).
(k) Power consumption.
(l) Any other observations that are required to verify a guaranteed quantity of cooling water, inlet and outlet temperatures of cooling water, *etc.*

Results derived from observations

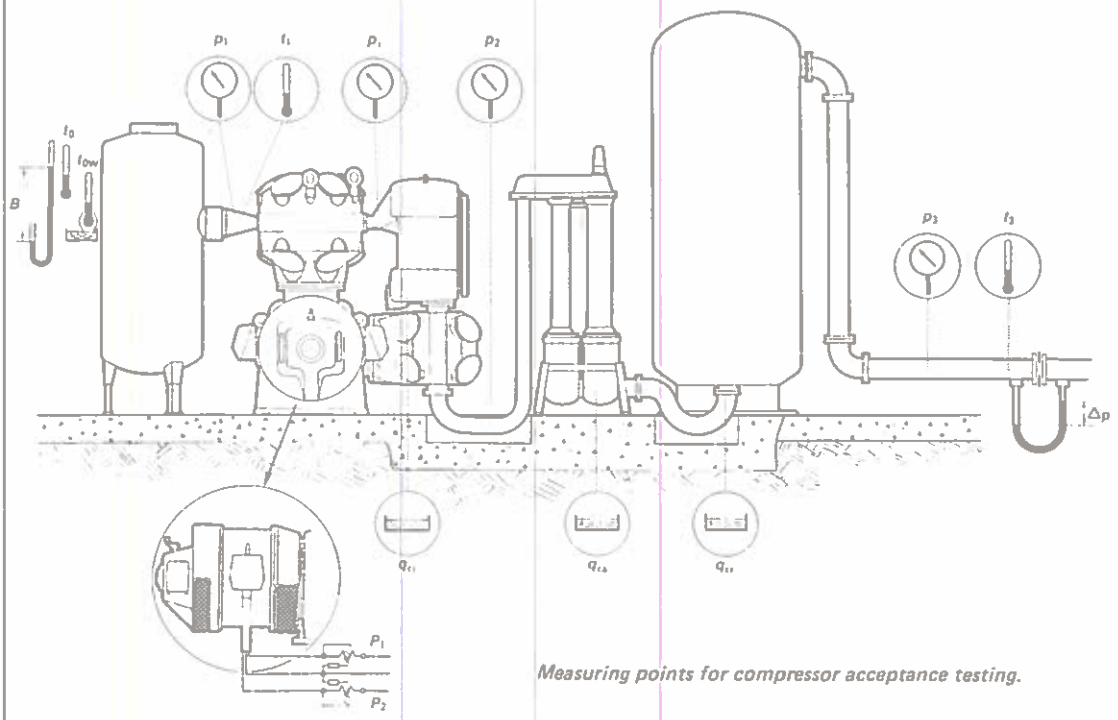
(a) Nozzle constant.
(b) Capacity (FAD).



Diagrammatic arrangement of test layout for rotary compressor.



Diagrammatic arrangement of test layout for reciprocating compressor (BS1571:1975).



Measuring points for compressor acceptance testing.

Adjustment of Results to Guaranteed Conditions

(a) Corrected capacity (FAD)

$$= \frac{\text{test FAD x guaranteed speed}}{\text{test speed}}$$

If the speed for a specified FAD is not guaranteed, the correction does not apply provided that the machine gives the specified FAD at a speed which does not adversely affect its service reliability (see also Table II).

(b) Pressure at delivery. For positive displacement compressors no correction is necessary as it can safely be assumed that the specified delivery pressure can be obtained with specified intake conditions.

(c) Pressure at intake. For exhausters no correction is necessary within the permissible deviation — see Table I.

(d) Corrected power consumption

= power consumption

$$\times \frac{\text{guaranteed absolute inlet pressure}}{\text{test absolute inlet pressure}} \times \frac{\text{guaranteed speed}}{\text{test speed}}$$

TABLE I – PERMISSIBLE DEVIATIONS FROM GUARANTEE CONDITIONS (BS1571)

Parameter	Deviation
Speed	±5%
Absolute intake pressure	±5%
Temperature of cooling water	±8%
Pressure ratio	±1%

TABLE II – TOLERANCE ON CAPACITY AND SPECIFIC ENERGY CONSUMPTION (BS1571)

Shaft Input at Normal Load	100% Capacity		90% Capacity		No Load Power
	Capacity	Specific Energy Consumption	Capacity	Specific Energy Consumption	
	%	%	%	%	%
below 10 kW	6	7	–	–	20
10 kW to 100 kW	5	6	7	7	20
above 100 kW	4	4	5	6	20

This is the only correction necessary provided that the deviations of test conditions from guarantee conditions do not exceed the values given in Table I.

See also chapter on *Air Flow Measurement*.

Compressor Selection

THE MAIN parameters involved in selecting a compressor for a particular application are pressure and air delivery required, initial cost and running costs.

As far as delivery pressure is concerned, all types — reciprocating, screw, vane and dynamic — can readily produce pressure up to 10 bar, the only exception being those types specifically classified as blowers. Where higher pressures are required a three stage *reciprocating compressor* becomes a logical choice; or a *centrifugal compressor* for larger capacities.

All types have overlapping performance envelopes for capacities up to about 300 m^3/min (10000 ft^3/min) except that efficiencies of centrifugal compressors may not be attractive below about 170 m^3/min (6000 ft^3/min). Capacities offered by centrifugal compressors extend up to about 2800–3500 m^3/min (100000–125000 ft^3/min), or well into the range also covered by *axial flow compressors*. Thus the two overlap in a strictly competitive range, although the axial flow type becomes increasingly attractive as the capacity required increases. Thus with very high capacities an axial flow compressor is the normal choice although the pressure range available may be more limited than with a centrifugal type. The lower limit of capacity at which the axial flow compressor usually ceases to be competitive is about 280 m^3/min (10000 ft^3/min).

At lower capacities vane compressors are best suited to low volume medium pressure duties, but may be directly competitive with reciprocating compressors up to, and well beyond, delivery of 100 m^3/min (3500 ft^3/min). Modern designs of screw compressors can also be directly competitive with both reciprocating compressors up to, and well beyond, delivery of 100 m^3/min (3500 ft^3/min); and turbo machines with deliveries up to 600 m^3/min.

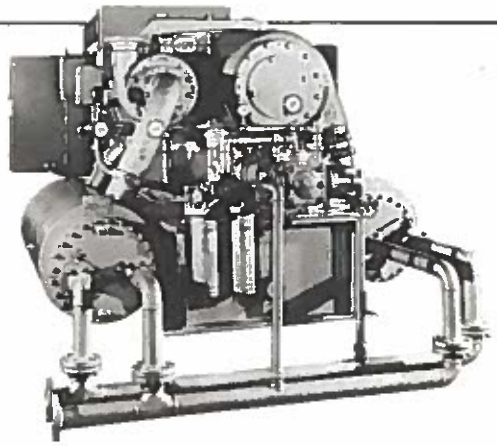
Generalized performance envelopes for different types of compressors are given in Figs 1 and 2, and some comparative compressor characteristics are summarized in Table I.

Costs (See also Figs 3 and 4)

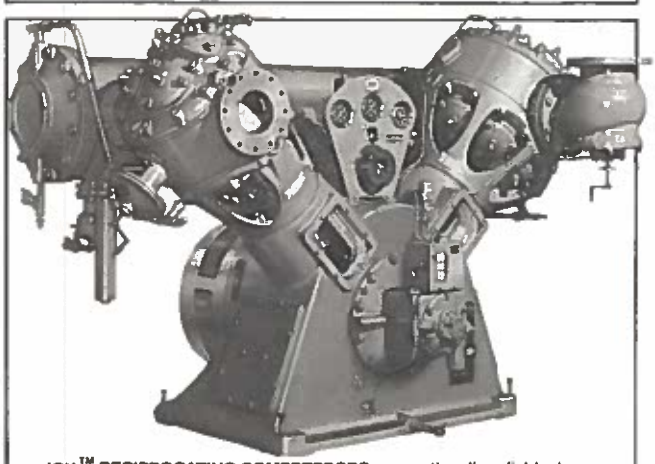
For compressor sizes below about 6 m^3/min (200 ft^3/min) capacity, running costs do not play a significant part in choice of compressor. Thus only initial cost is a primary parameter where the demand is relatively small and/or intermittent. This would generally favour a reciprocating machine. Compressor efficiency may also be insignificant, particularly if use is intermittent and off-loading is incorporated. Where the demand is high and continuous, overall efficiency may be more significant than initial cost.

Your quick reference guide to energy-saving air compressors

TWISTAIR™ oil-injected, rotary screw compressor. Quickly installed, quiet in operation and virtually maintenance-free. A complete plant air 'package' with 18 models covering a range from 2.6 to 21m³/min. (90 to 740cfm) with choice of pressures 7.5 or 10 bar (110 or 145psig).



TURBO AIR™ **COMPRESSORS** - Packaged centrifugal compressors of intergral geared, multi-stage design; these machines produce oil-free air on continuous operation for plant and process application. Compact and easily installed, the range covers 2380 to 66300m³/hr (1400 to 39000) cfm at pressures to above 50 bar (750 psig).



JOY™ **RECIPROCATING COMPRESSORS** - exceptionally reliable, heavy-duty water-cooled machines of 'oil-free' or lubricated construction. Single and two-stage machines cover capacities from 2.3 to 31.1m³/min. (80 to 1100cfm) at normal working pressure of 7 bar (100psig).

Joy capability in air compressors meets the needs of all manufacturing and process industries for intermittent or continuous duty. Quiet operation, energy saving control systems; low maintenance requirements; wide choice of oil-free models...and many more benefits in a range that you should know more about. Write for more details to:

JOY MANUFACTURING COMPANY (UK) LIMITED
Compressor Division, Peel Park Place,
East Kilbride, Glasgow G74 5LN.
Telephone East Kilbride 39121. Telex 77368

unlimited air compressed into a single name

JOY™

T M - Trade mark of Joy Manufacturing Company

CompAir

ressed

The power to boost industry.

CompAir is compressed air.

And at CompAir we've been an integral part of the technological advances in industry for well over eighty years.

Today, our products are relied upon in over 100 countries and as many different industries again; we have a breadth and depth of experience that no other compressor manufacturer can match.

And now we boast a wider range of compressors and ancillary equipment than ever before.

Covering pressures from 11 psi.g. to 6000 psi.g, delivering from 2 cfm to 60,000 cfm, it's a range that's technically the most advanced you are likely to find anywhere.

Built to withstand the most extreme industrial conditions, and backed by our unrivalled nationwide sales and service network, only compressors with the CompAir badge have the pedigree that modern industry demands.

Because only they have the power to boost industry.

Simply tell us which industry you're in by placing a tick in the appropriate box and we'll send you some useful cost effective facts about compressed air.

Food, Drink & Tobacco	☐	Metal Goods	☐
Coal & Petroleum Products	☐	Textiles	☐
Chemical & Allied Industries	☐	Leather Goods	☐
Metal Manufacture	☐	Clothing & Footwear	☐
Mechanical Engineering	☐	Brick, Pottery Glass Cement	☐
Instrument Engineering	☐	Timber & Furniture	☐
Electrical Engineering	☐	Paper Print & Publishing	☐
Shipbuilding	☐	Other Manufacturing	☐
Vehicles	☐		

Name_____

Position_____

Company_____

Address_____

_____

_____ Tel: _____

CompAir BroomWade

Air power to boost industry.

CompAir Industrial, BroomWade Works, High Wycombe, Bucks. HP13 5SF. Tel: (0494) 21181 Telex: 837371

PH/82

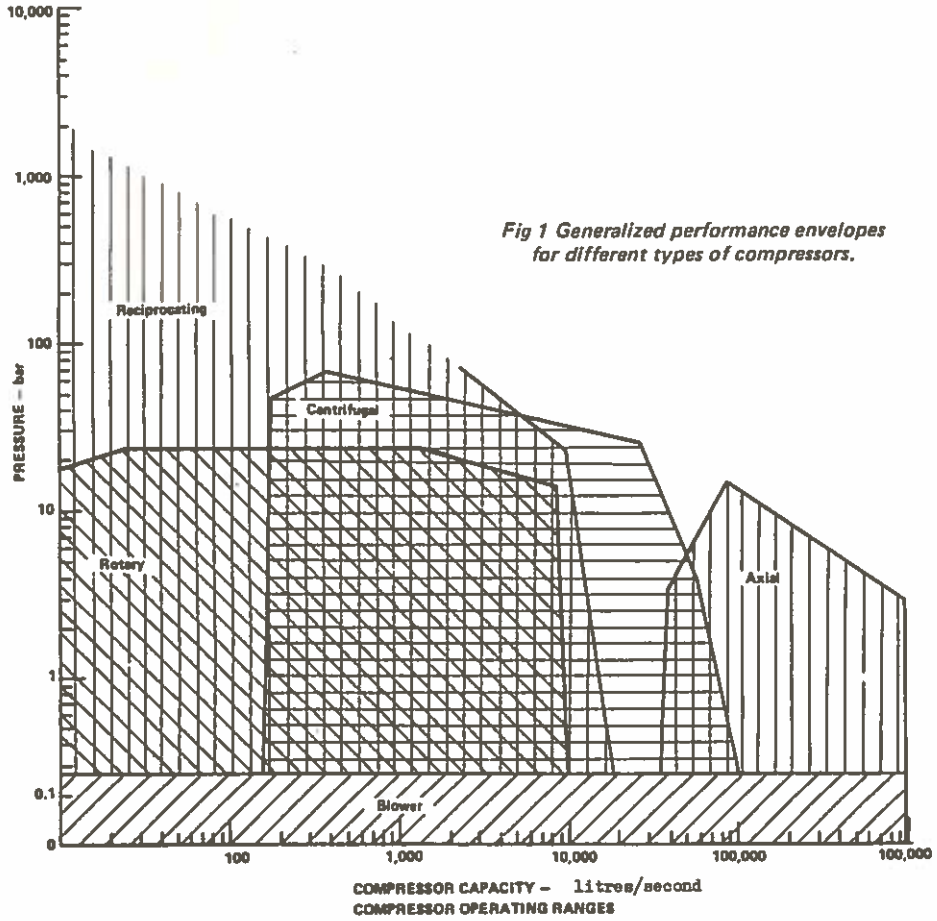


Fig 1 Generalized performance envelopes for different types of compressors.

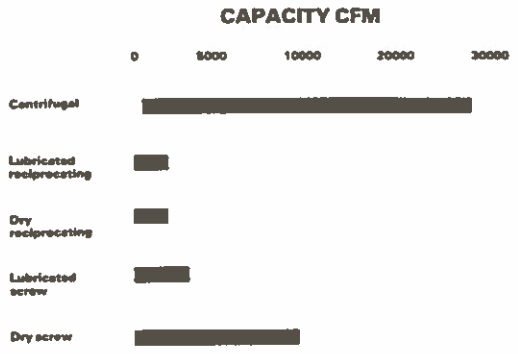


Fig 2 Capacity ranges for different types of compressors.

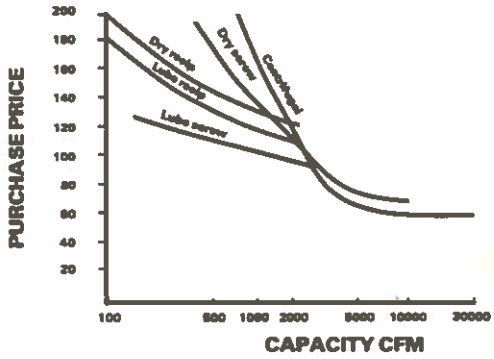


Fig 3 Comparative costs of compressors.

TABLE I – COMPRESSOR CHARACTERISTICS – PRESSURE, CAPACITY AND SPEED

Type	Sub-Type	Pressure		Delivery		Speed	Remarks
		bar	lb/in^2	m^3/min	ft^3/min (FAD)	rev/min	
Reciprocating	Single-stage	up to 7	up to 100	below 8.5	below 300		Common choice for capacities up to 100 m^3/min
	Two-stage	up to 7	up to 100	over 8.5	over 300		
	Multi-stage	up to 10	up to 150 or higher	over 8.5	over 300		
Vane	Single-stage	up to 3.5	up to 50–60	1.5–1 500	50–50 000	250–3500 depending on size up to 3000	Water cooled above 2 bar
	Two-stage	up to 10	up to 120–150				
	Oil-cooled	up to 8	up to 120				
Roots	Single-stage	up to 1	up to 15	up to 1 500	up to 50 000	250–6 000 depending on size	Blowers
	Two-stage	up to 2	up to 30				
Screw (dry)	Single-stage	up to 4	up to 60	17–550	600–20 000	300–25 000	Essentially high speed machines
	Two-stage	up to 10 or higher	up to 150 or higher				
		up to 30	up to 450				
Screw (oil-filled)	Single-stage	up to 7	up to 100	up to 85	up to 3 000		Highly competitive type in size ranges available
Centrifugal	Single-stage	0.5	6	6.5 upwards	230 upwards	up to 25 000	Blowers
	Four-stage	2.0	30	up to 3 500	up to 125 000		Water-cooled
	Five-stage	2.0	35	170–3 000	6 000–100 000		
	Multi-stage	7.0	100				
	Multi-stage in series	above 2,0	above 35	up to 3 000	up to 100 000		Compression ratio 1.5 to 1.6 per stage
Diaphragm	Single-stage	4	60	up to 0.7	up to 25		
	Two-stage	7	100	up to 0.2	up to 6		
	Three-stage		Higher pressures when hydraulically actuated				
Liquid ring	Single-stage	up to 1.5	up to 20	up to 150	up to 5 000	up to 3 500	Large volumes at low pressure or low volumes at higher pressures
	Two-stage	up to 3	up to 30–45				

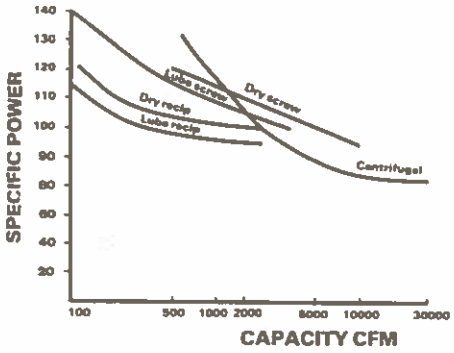


Fig 4 Comparative efficiencies of compressors.

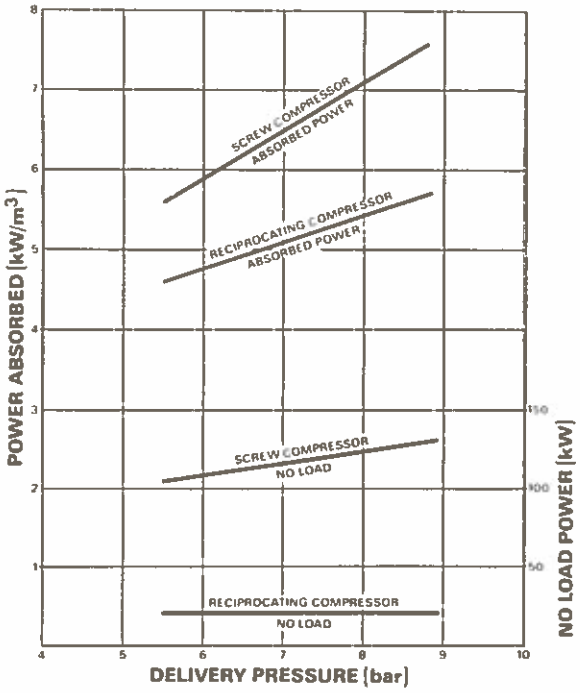


Fig 5 Graph showing the respective power absorbed and the no-load power requirements of a ·28 m^3 /min (1000 ft^3 /min) two stage reciprocating compressor and an equivalent oil-flooded screw compressor.

Efficiency figures quoted arbitrarily can be misleading. With good design, most compressor types should prove capable of achieving overall efficiencies of between 70% and 85%, relative to the theoretical power required for ideal isentropic compression with no friction, fluid or other mechanical losses. Some types of compressor are inherently less efficient than others, however, and all are influenced by machine size, speed and other operating parameters. Thus, where overall efficiency is a major parameter and thus exact determination of efficiency is required, this is normally given as a guaranteed performance.

Efficiency also depends on cooling and intercooling. Good cooling system design reduces water cooling requirements and cost. Fan cooling should be cheaper, but tends to be rather less efficient and more restricted in application.

It is difficult to compare the efficiencies of different types of compressors offering similar pressure and capacity except on purely empirical grounds. An identical comparison of power consumption under no load conditions between a reciprocating compressor and a screw compressor is not possible, for example, since the former is normally fitted with suction valve lifters and the

latter equipped with throttle control. They can only be assessed on comparative performance and specific energy requirements. Some typical figures are:-

Capacity	Type of Compressor		
	Reciprocating (piston)	Screw	Vane
2.8 m^3/min (100 ft^3/min)	23.5	26.1	26.3
14 m^3/min (500 ft^3/min)	20.1	25.0	27.0
28 m^3/min (1000 ft^3/min)	19.0	22.4	–
85 m^3/min (3000 ft^3/min)	18.2	21.3	–

Fig 5 shows a direct comparison between a two-stage reciprocating compressor with a capacity of 28 m^3/min (1000 ft^3/min) and an equivalent oil-flooded screw compressor. An analysis of running costs on this basis is:-

	Reciprocating Compressor	Oil-flooded Screw Compressor
Power absorbed at 28 m^3/min and 7 bar	142 kW	179.8 kW
Power consumed at 92% electric motor efficiency	176.3 kW	223.4 kW
Cost for 3000 hours at full load at 0.02 per unit	£10578	£13404
Power consumed at no-load	80 kW	120 kW
Cost for 1000 hours at no-load	£1600	£2400
Total cost for 4000 hours at 74% weightup	£12178	£19804
Difference		+ £7626
Difference per hour		+ £1.906

This, however, is not a complete picture since it concentrates entirely on running costs. These are a *major* cost (and likely to rise with time). Other costs involved are:-

(i) *Mounting requirements* — which can vary with the type and size of compressor. The cost of suitable foundations is also substantially higher with some types than others.

(ii) *Driver cost* — again variable with type and size of machine. Optimum compressor speed is also significant here, for this governs the suitability for direct drive, or the necessity for gearing up or down.

(iii) *Depreciation and maintenance* — which can be major overall cost factors after the first year.

Comparative features are summarized in Tables II and III.

TABLE II – COMPRESSOR CHARACTERISTICS INFLUENCING SELECTION

	First Cost	Efficiency	Installation	Vibration	Noise Level	Remarks
Reciprocating	Low to moderate	High		Reduced with L & W layouts	Very high	Lubricated and dry types
Vane	Low to moderate	Slightly inferior to reciprocators	Simple	None	Low (especially water cooled type)	Best suited to medium pressure and low volume duties
Screw	High	High (directly comparable with reciprocators)			Very high	Modern oil-filled designs can be competitive with reciprocating compressors and turbo machines
Centrifugal	High	Poor in small sizes; high in larger sizes	Requires suitable bed	Balance very important due to high speeds	Can be low with filtering	Essentially high speed machines for efficient performance. Not generally competitive below 170–200 m^3/min (6 000 ft^3/min)
Axial	High	High	Requires suitable bed		Can be low with filtering	Unrivalled performance for high outputs. Not generally competitive below 280 m^3/min (10 000 ft^3/min)
Diaphragm	Low	High		High	High	Limited capacity but particularly suited to handling toxic or corrosive gases
Roots	Low	Low but improves with increasing size	Simple	Low if rotors properly balanced	High	Most suited to low pressure (blower) high volume requirements

The general rule commonly adopted for assessing compressor life is ten years or 100000 hours, although this may be reduced substantially where the compressor is working under particularly adverse conditions. This can be taken as a typical figure from which to calculate a realistic estimate for depreciation. The figure of 100000 hours can be used for a compressor which is run continuously, and ten years for a compressor run intermittently; the lower hours in this case are offset by the deterioration likely to occur during periods of idleness.

Working time between overhauls or replacement can be considerably shorter, although logically all critical parts of a compressor should be designed for the same 10 year/100000 hour life. Certain types, however, may have parts with an inherently shorter life — *eg* the life of the blades on a vane compressor may be of the order of one year only, or even less. Also, of course, reliability, as expressed in terms of continued troublefree operation, can be a characteristic of the individual design and quality of construction.

A breakdown of *time* costs is then likely to be of the following order:

Capital costs spread over 10 years	25%
Power requirements	60%
Maintenance costs	15%
Total	100%

A further comparison between a reciprocating compressor and an oil-flooded rotary compressor taking maintenance costs into consideration is given below:

Conditions: 3000 operating hours/year; 75% load factor; £0.02/kWh; £8 hr labour costs.

	COMPRESSORS	
	Two-stage, air-cooled reciprocating compressor with air-cooled after-cooler/moisture separator	Single-stage, air-cooled, oil-injected rotary screw compressor with integral air-cooled aftercooler and moisture separator
Output @ 102 lb/in^{2}g	212 ft^3/min	220 ft^3/min
Total power input at full output	36.3 kW	43.8 kW
Input at no load	7.4 kW	12.6 kW
On-load power costs	£1856.25	£2158.25
Off-load power costs	£133.73	£252.57
Total power costs	£1989.98	£2410.82
Maintenance costs — labour	£138.00	£65.00
Maintenance costs — parts	£901.50	£103.17
Total maintenance costs	£1039.05	£168.17
Power + maintenance costs	£3029.48	£2578.99
Difference	+ £405.49	
Difference per hour	+ £0.15	

Note: Due to its higher output, the rotary-screw compressor has a lower load factor than the reciprocating compressor, *ie* 72.27% of that of the reciprocating compressor.

Saving Power

Considerable power can be wasted if an oil-flooded single stage rotary screw compressor is operating on a no-load basis for any length of time. Obviously, it is as important to avoid wasting air when the machine is operating for no purpose as it is to avoid wastage through leakage.

By using a compressor system with a modulating or throttle inlet valve to control the airflow through the machine to match the demands of the air system, there is very little saving of power when the compressor is off-load. This is because at half capacity the power consumption still represents 80% of the full-load power – assuming there is no air receiver.

The full-load/no-load system using an air receiver may be slightly more expensive on first cost to purchase but it saves considerably on power costs during the 'no-load' times. In this system the compressor is at full capacity until the air system pressure reaches a pre-set unloading level. The machine will then unload by closing the inlet valve. Since no air is needed, the system pressure will drop until the preset loading pressure is reached, at which point the air will build up again. Power consumption at no-load is approximately 25% of the full-load power because the compressor is still working against a minimum pressure.

Even more remarkable is the case of reciprocating piston machines using an air receiver. Here the unloaded power consumption is even lower – a mere 8% of load power.

Savings can be made if the right compressor is used for the right application. Where power represents some 60% of the current operating costs, efficiency of the complete compressor relates far more to these costs than it does to the misleading and 'non-cost related' term, volumetric efficiency.

Vibration

Rotary machines are inherently smoother-running than reciprocating machines. The latter may require particular attention to prevent excessive vibration (and noise) being generated, *eg* by the application of anti-vibration mounts so that the machine is suitably isolated. In the case of rotary machines, the vibration level depends primarily on the degree of dynamic imbalance present in the rotating elements. Effective vibration damping can add appreciably to the total cost of installation.

Noise

Some types of compressor tend to be inherently noisier than others although a general description of this particular parameter is impossible, particularly as the noise spectra generated by different machines may vary appreciably.

With some types air noise can be reduced substantially by simple treatment, *eg* with a filter type silencer applied at a suitable point. With others this may not be possible, except by isolating the machine in a complete enclosure. 'Machine' noise transmitted through the casing may also vary appreciably in level and spectra with different types.

The subject of noise (and vibration) control is a highly specialized one, requiring knowledgeable technical treatment to achieve any realistic improvements.

Lubrication

Lubrication is seldom a problem with modern compressor designs. Pressure lubrication is now generally employed on modern reciprocating compressors. Some types can run dry and would be a normal choice where 'oil free' compressed air is required, (see Table III).

TABLE III – COMPARISON OF COMPRESSOR TYPE FEATURES*

Compressor Characteristics	Lubricated Reciprocating	Dry Reciprocating	Lubricated Screw	Dry Screw	Centrifugal
1. Pressure volume selection	See Fig 2				
2. Low purchase price	See Fig 3				
3. High initial efficiency	See Fig 4				
4. High life expectancy	7	2	7	3	10
5. Oil-free air	0	10	0	10	10
6. Low installation cost	5	5	10	9	10
7. Low foundation cost	4	4	10	10	10
8. High long term efficiency	6	3	7	5	10
9. High part load efficiency	7	7	2	2	9
10. Low regular maintenance	4	2	8	6	10
11. Ease of inspection	6	5	5	3	10
12. Quietness of operation	5	5	6	6	9
13. Freedom from vibration	3	3	10	10	10
14. Safety	6	8	6	8	10
15. Capability of uprate	0	0	0	0	10
	Gradings against items 4–15 are a guide to capabilities on a scale of 0–10				

*United Technologies Elliott

Valve Design

Good valve design ensures low-loss performance and reliability. This can be a particularly important factor in the case of reciprocating compressors.

Control

The most common form of control is *discontinuous regulation.* The degree of control possible depends on the type of compressor and may influence the choice of compressor type in some applications. The ability to match a compressor with associated plant may also be a consideration.

See also chapters on *Compressors, Compressor Controls* and *Compression Lubrication.*

SECTION 2b

Air Lines

Compressed Air Mains

ALTERNATIVES AVAILABLE are a *dead-end main* or a *ring main.* A dead-end main has the advantage of distributing the compressed air over the shortest possible run, reducing pressure drop and economizing on pipe cost. Its chief disadvantage is its lack of flexibility, although the service supplied by the compressor can be extended by the use of a split flow main — taking two separate mains from the compressor. Typical dead-end mains systems are shown in Fig 1.

Fig 1

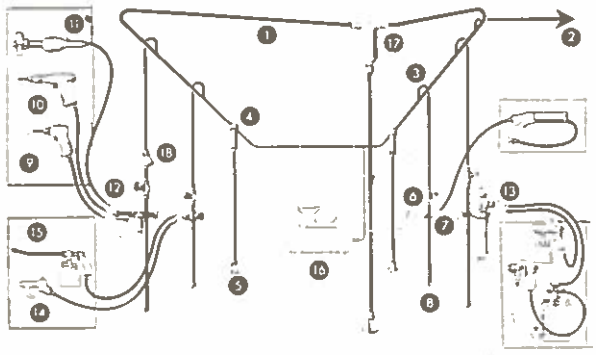


Fig 2 Ring-main system (Broom & Wade)

1 – ring main
2 – branch main
3 – take-off points
4 – filter/separator in main
5 – automatic drain
6 – isolating valves
7 – connecting point
8 – drain cock
9,10,11 – tools
12 – separator/lubricator
13 – separator/reducing valve
14 – blow gun
15 – oil/water jet
16 – compressor
17 – moisture trap
18 – solids separator

A ring main has the advantage that being in the form of a 'closed circuit', the velocity of air in the main will be reduced and the pressure drop should be less. The ring main itself may be of appreciable volume and capable of acting as a form of 'reservoir' to assist in maintaining pressure

at any particular point of use where there is a sudden draw-off. Also, if the system is extended by the addition of another branch, this branch can draw its supply from both ends of the ring main, reducing air velocity and minimizing pressure drop. A ring main system thus offers maximum flexibility with functional advantages and is generally the preferred choice — Fig 2.

Regardless of the type of main, the most important consideration is that the pipeline should be of suitable size to pass the maximum volume of air likely to be required by the system, without any significant pressure drop. Pressure drop represents a power loss, not just in terms of output power at the take-off points, but also input power required (and consequently higher operating costs) to maintain the desired working level. A further advantage of a low pressure drop is that the operating pressure range of the compressor can be significantly reduced.

Mains Sizing

Theoretically, at least, pressure drop would be reduced to a minimal figure by the selection of a very large diameter pipe. This, however, is not necessary or even practical. Excessively oversize pipes can cause air (and time) to be wasted in filling the extra volume, although actual losses in this respect can be relatively insignificant. Saving unnecessary extra cost of pipelines is more important.

As a general design rule a pressure drop of the order of 5% per 30 m (100 ft) of pipe is satisfactory, bearing in mind that actual pressure drop is also dependent on air velocity through the pipe which in turn is governed by demand. If further extension of the main is likely, then 'oversizing' by about 50% on the original figure should be considered.

Mains sizing can be determined directly from theoretical or empirical data, or based on accepted recommendations. 'Maximum' flow rates can be quoted for specific pipe sizes, established on the basis that this maximum rate corresponds to a pressure drop of 10% — see Table I.

TABLE I — MAXIMUM RECOMMENDED FLOW THROUGH MAIN LINES

Nominal Bore (mm)	Rate of Air-flow at 7 bar (litres/second)
6	1
8	3
10	5
15	10
20	17
25	25
32	50
40	65
50	100
65	180
80	240
100	410
125	610
150	900

Note: Velocity of air-flow less than 6 m/s

For continuous flow it would then be advisable to base the pipe size on a 75% flow rating. Further, the continuous flow figure should be based on the maximum continuous flow demand

TABLE II – VARIATION OF PRESSURE DROP WITH FLOW RATE

Design rating for continuous flow ... ... ↓

Flow Rate %	50	60	70	75	80	90	100	110	120	150
Pressure Drop lb/in^2	2.5	3.6	4.9	5.6	6.4	8.1	10	12.1	14.4	22.5
Pressure Drop bar	0.2	0.25	0.35	0.40	0.45	0.57	0.7	0.85	1.0	1.6

Design maximum permissible flow rating ... ... ↑

TABLE IIIA – MAXIMUM RECOMMENDED FLOW THROUGH ABS PIPE

10 BAR PIPE								
Applied Pressure	0.6 bar	1.0 bar	1.6 bar	2.5 bar	4.0 bar	6.0 bar	7.0 bar	10.0 bar
Pipe Size (o.d.) mm								
12	0.60	0.95	1.52	2.41	4.02	6.25	7.5	11.1
16	1.35	2.35	4.0	6.4	10.9	16.9	19.7	28.8
20	1.80	3.3	5.5	8.6	14.0	21.8	25.7	38.5
25	3.25	6	10.0	16.5	27.5	42.5	50.3	74.5
32	5.5	10	16.5	27.5	45.5	71.5	85	127.5
50	20	36	60	97	163	250	294	425
63	36	64	107	176	290	450	528	760
75	55	105	175	285	475	750	875	1270
90	95	170	280	455	745	1145	1360	1965
110	165	280	465	740	1240	1930	2280	3310

Note: Flow in litres per second

TABLE IIIB

12.5 BAR PIPE								
Applied Pressure	0.6 bar	1.0 bar	1.6 bar	2.5 bar	4.0 bar	6.0 bar	7.0 bar	10.0 bar
Pipe Size (o.d.) mm								
12	0.45	0.8	1.30	2.10	3.50	5.55	6.6	9.80
16	1.05	1.8	2.85	4.65	7.75	12.20	14.6	21.60
20	1.35	2.3	3.7	6.0	9.95	15.75	18.8	27.85
25	2.2	4.0	6.8	11.3	19.2	31.0	36.5	54
32	4.5	9.0	15.5	25.3	41.8	63.5	74.5	107
50	16.5	27.5	44	71	122	193	229	337
63	30	50	83	132	229	366	434	646
75	45	80	137	220	360	565	665	975
90	75	135	225	350	570	885	1040	1525
110	135	225	380	600	1000	1570	1870	2710

Note: Flow in litres per second

and not on the average, to allow for the possibility of all services fed by the main being used simultaneously. Table II shows how departures from design flow rate can affect the pressure drop, calculated for a 7 bar (100 lb/in^2) supply pressure.

More specific figures are quoted as recommendations for ABS pipes — see Tables IIIA and IIIB.

The maximum flow rate required can be based on 1.5 to 3.0 times the average flow, depending on the number of outlets in use at any one time and the particular geometry of the system.

A specific maximum pressure drop figure may be adopted to cover the requirements of the system. The choice of this figure is somewhat arbitrary. A compressor is normally capable of supplying a higher pressure than that needed for efficient operation of the various devices it feeds, but against this is the fact that pressure drop represents lost work or wasted power input. The system designer should aim at achieving a pressure drop of no more than 3–8%, as representing efficient working. In practice a higher figure is perfectly acceptable, and more realistic requirements are:

(i) The lowest practical pressure drop in the main distribution system (*eg* in a ring main).

(ii) A maximum of 10% pressure drop at the farthest point in the system in the device itself, although in particular circumstances a higher figure of up to 15% pressure drop may well be tolerated at such a point.

It should also be appreciated that the geometry of the mains has a bearing on actual pressure drop, *ie* bends and/or fittings represent additional frictional losses and thus pressure drop. Here generous radius bends in the pipe itself are more favourable than bends achieved via fittings. Examples of pressure drop through fittings are given in the chapter on *Pressure Drop Calculations.*

Basic Geometry

Overhead location of an air main is the best choice, whenever this is feasible, particularly with ring mains. This enables the mains to cover the working area(s) in a direct manner. Besides being cheaper and simpler, this can also minimize the number of bends required.

In the case of a dead-end mains there is often a case for adopting a low level system — *eg* see Fig 3. This can offer particular advantages for factory installations where each main is supplied by its own compressor. The main is run at a height of 0.45–0.60 m between back-to-back tool stations or under a bench run, and carried on floor-mounted stanchions as necessary. A spacing of 3–4 m apart is generally satisfactory.

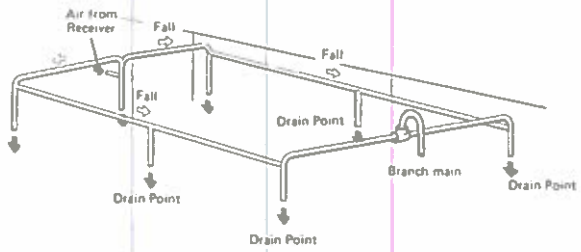
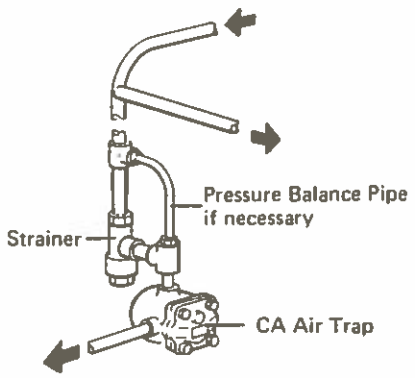


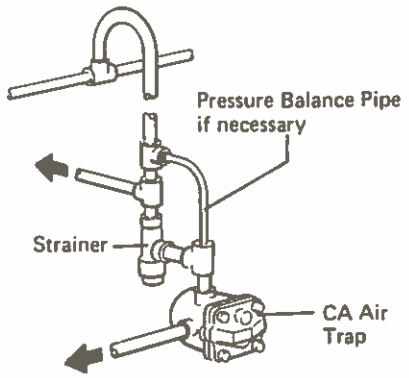
Fig 3

Mains Fall

It is particularly important that an air main should be given a fall in the direction of the airflow (Figs 4 and 5). A fall of 1 in 100 or 1 in 120 is generally enough to ensure that water separating out is carried along to a low point by both gravity and airflow. Moisture traps and/or drains can then be installed at suitable points along the run and at each individual low point. Automatic traps or drains are preferred as these cannot become overfilled and thus inoperative.



'Spirax' Air Trap draining a relay point



'Spirax' Air Trap draining a branch main

Fig 4

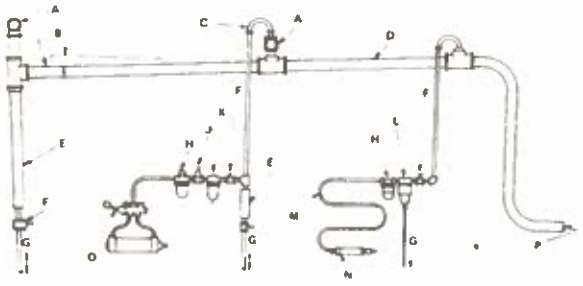


Fig 5 Dead-end main (Maxam).

A – branch main
B – pitch with flow
C – wide pattern return bend
D – main
E – trap or water leg
F – stop valve
G – drain
H – lubricator
J – regulator
K – filter
L – automatic drain air filter
M – flexible hose
N – air drill
O – hand-operated valve and air cylinder
P – from receiver

Branch mains or individual take-off points should emerge in the form of an upward loop. This will help contain any condensate present in the ring main and minimize the carry over of moisture to tool points, *etc*. A filter/separator should be fitted in the ring main at a high point between the compressor and the first take-off point, to reduce further the moisture entering the main and branch lines. This should be of the automatic type.

Each branch and take-off point should have its own isolating stop valve and drain or drain cock at the blind end. Filters, separators and lubricators should be in the individual lines feeding tools or appliances requiring protection and/or lubrication. The choice of these devices will depend on the type of tool or appliance concerned. Tools normally require a moisture separator/lubricator

in the feed line. For paint spraying a moisture separator/reducing valve unit is required, the latter enabling the working pressure to be controlled at the take-off point. Blow guns should also be used via a pressure-reducing valve to control the force of the air jet.

Where the main involves any appreciable length of run, flange connections and expansion joints should be used throughout the system as necessary. The run should also be above ground and accessible so that any leaks can readily be detected and rectified.

Pressure drop is adversely affected by restrictions in the flow. Thus the interior of all pipes should be smooth and clean, and bends should be gradual with generous radii. Sharp elbows, tees and similar fittings should be avoided and all fittings used should be full-bore. Similar considerations apply to valve and hose connections.

Adequate pipe support is necessary for ring mains. As a general guide, the following recommendations for support spacing apply, (but see also Table IV):

Main diameter		Pipe run between supports	
mm	in	metres	feet
25	1	2.4	8
32	1¼	2.4	8
38	1½	2.4	8
51	2	3.0	10
76	3	3.4	11
102	4	3.7	12
127	5	4.0	13
152	6	4.3	14

TABLE IV — INTERVALS BETWEEN PIPE SUPPORTS

Steel Pipework			ABS Pipework	
Pipe size Nominal bore mm	Maximum intervals for vertical runs m	Maximum intervals for horizontal runs m	Pipe Outside diameter mm	Support Spacing at 20°C m
8	1.25	1.00	16	1.10
10	1.25	1.00	20	1.25
15	1.75	1.25	25	1.50
20	2.70	1.75	32	1.75
25	2.50	1.75	50	2.0
32	3.00	2.50	63	2.3
40	3.00	2.50	75	2.6
50	3.00	2.75	90	2.9
65	3.50	3.00	110	3.2
100	3.50	3.00		
150	4.25	3.50		
200	4.50	3.50		
250	5.18	4.25		
300	5.48	4.87		
over 300	5.48	4.87		

For each 10°C temperature rise, reduce support spacing by 10%.
These support centres can be doubled for vertical pipes.

Outdoor Air Lines

Outdoor air lines present special problems — notably the need to protect the outside of pipes with an anti-corrosive paint or similar treatment, and the desirability of burying the lines in the ground at frost free depth to prevent freezing of condensate and pipe obstruction or blockage developing at low ambient temperatures. Underground installation can be costly, particularly as inspection wells are also necessary to gain access to condensate traps and valves.

Overground installation, with the pipes suspended on posts or supported on plinths is generally cheaper and simpler. Quick-coupling pipes are recommended in this case. This form of joint has an expansion-contraction capability to counteract both thermal movements and some misalignment. With overground lines, however, particular attention must be given to condensate separation where there is a risk of freezing.

It is also recommended that all outdoor pipes should be of welded steel construction, hot-galvanized and fitted with flanged joints for water traps and valves.

Leakage Losses

Any leakage from an air main (or any other part of the system) can result in substantial power losses, *eg* at a typical system pressure of 7 bar, the leakage through a hole of approximately 1.6 mm can represent an additional 1 kW of power *per second* demanded from the compressor to make up.

In practice some leakage is to be regarded as inevitable and accommodated by adopting a nominal figure for leakage of 5% of the total compressor capacity. Higher leakage rates are, unfortunately, commonplace in systems where maintenance is poor, and in some cases may exceed 25–30% of the compressor capacity. A satisfactory maintenance programme should thus aim at holding leakage to acceptable levels, the most common source of loss being leakage at hoses, couplings and valves.

Two standard methods of determining the leakage rate in a complete system are:

(i) First determine the volume (V) (litres) of the air mains downstream of the receiver isolating valve including all branch and drop pipes. Pump up the whole system to the normal operating pressure (P_1) and then shut down the compressor. Close the receiver valve and make certain that all tools and equipment are isolated. Check the time for the system to leak down to a lower pressure (P_2). $P_1 - P_2$ is thus the pressure drop. The amount of leakage in litres per second of free air is computed as follows:

$$\frac{V(\text{litres}) \times (P_1 - P_2)\,(\text{bar})}{t\,(\text{secs})}$$

Example:

$V = 300$ litres
$P_1 = 7$ bar
$P_2 = 6$ bar
$t = 300$ seconds

$$\frac{300 \times (7 - 6)}{300} = 1 \text{ litre/second}$$

(ii) The second method is by using a compressor of known capacity as a measuring device. This machine should be run at full load with all equipment isolated. When the system is charged to operating pressure, the compressor will unload. Due to air leaks the air pressure in the system will fall, causing the compressor to come on load. The duration of the period (in seconds) for which the compressor is 'on-load' (T) and 'off-load' (t) should both be recorded at least four times. If the compressor and capacity is Q litres per second, and the loaded time is T seconds, the amount of air delivered is Q x T litres. If the total running time is t + T seconds, the average volume of air leakage is:

$$\frac{Q \times T}{t + T} \text{ litres per second}$$

See also chapters on *Branch Lines*.

Branch Lines

BRANCH LINES are normally of relatively small diameter compared with the air mains, but may range up to 80 mm diameter (or even 100 mm diameter in the case of hose). Pressure drop can be calculated from first principles (taking the losses through fittings, *etc* into account), or line sizing based on recommended practice – see Tables I, II and III.

For rigid lines, steel or copper tubes are used, and to a more limited extent ABS pipes. Flexible tube materials may be preferred at specific locations, *eg* take-off points. These may be of rubber, nylon, PVC or other plastics. Nylon tubing is generally suitable for working pressures up to a maximum of 35 bar (500 lb/in^2) at room temperature, depending on size. It is also produced in the

TABLE I – MAXIMUM RECOMMENDED FLOW THROUGH BRANCH LINES OF STEEL PIPE
Maximum recommended air flow (litres/second Free Air)† through medium series steel pipe for branch mains not exceeding 30 metres length (ISO 65).

Applied Gauge Pressure bar	NOMINAL STANDARD PIPE SIZE (NOMINAL BORE) – MILLIMETRES										
	6 mm	8 mm	10 mm	15 mm	20 mm	25 mm	32 mm	40 mm	50 mm	65 mm	80 mm
0.4	0.3	0.6	1.4	2.6	4	7	15	25	45	69	120
0.63	0.4	0.9	1.9	3.5	5	10	20	30	60	90	160
1.0	0.5	1.2	2.8	4.9	7	14	28	45	80	130	230
1.6	0.8	1.7	3.8	7.1	11	20	40	60	120	185	330
2.5	1.1	2.5	5.5	10.2	15	28	57	85	170	265	470
4.0	1.7	3.7	8.3	15.4	23	44	89	135	260	410	725
6.3	2.5	5.7	12.6	23.4	35	65	133	200	390	620	1085
8.0	3.1	7.1	15.8	29.3	44	83	168	255	490	780	1375
10.0	3.9	8.8	19.5	36.2	54	102	208	315	605	965	1695
12.5	4.8	10.9	24.1	44.8	67	127	258	390	755	1195	2110
16.0	6.1	13.8	30.6	56.8	85	160	327	495	955	1515	2665
20.0	7.6	17.1	38.0	70.6	105	199	406	615	1185	1880	3315

General Notes:
† The flow values are based on a pressure drop as follows:
10% of applied pressure per 30 metres of pipe of 6–15 mm nominal bore.
5% of applied pressure per 30 metres of pipe of 20–80 mm nominal bore.

TABLE II – MAXIMUM RECOMMENDED AIR FLOW (LITRES/SECOND)† THROUGH COPPER OR NYLON TUBING AND CORRESPONDING PORT THREAD (ISO 1179)

Applied Gauge Pressure bar	NOMINAL TUBING SIZE (OUTSIDE DIAMETER) – MILLIMETRES									
	4 mm G1/16	5 mm G1/8	6 mm G1/8	8 mm G1/4	10 mm G1/4	12 mm G3/8	16 mm G1/2	18 mm G1/2	22 mm G3/4	28 mm G1
1.6	0.05	0.12	0.19	0.48	0.9	1.8	4.2	6.5	10.0	18.0
2.5	0.08	0.18	0.26	0.70	1.3	2.8	7.2	8.8	14.0	30.0
4.0	0.14	0.28	0.45	1.2	2.2	4.5	12.0	15.0	24.0	50.0
6.3	0.22	0.48	0.72	2.0	3.8	7.6	20.0	25.0	40.0	80.0
8.0	0.28	0.62	0.95	2.6	4.8	9.5	26.0	31.0	52.0	110.0
10.0	0.36	0.81	1.3	3.5	6.2	12.0	34.0	41.0	68.0	140.0
12.5	0.46	1.1	1.7	4.3	8.0	17.0	44.0	53.0	85.0	180.0

Notes: † The flow rates are selected to provide an approximate pressure drop with either nylon or copper tubing as follows:

15% of applied pressure per 30 metres of tubing in sizes 4–16 mm inclusive.
10% of applied pressure per 30 metres of tubing in sizes 18–28 mm inclusive.

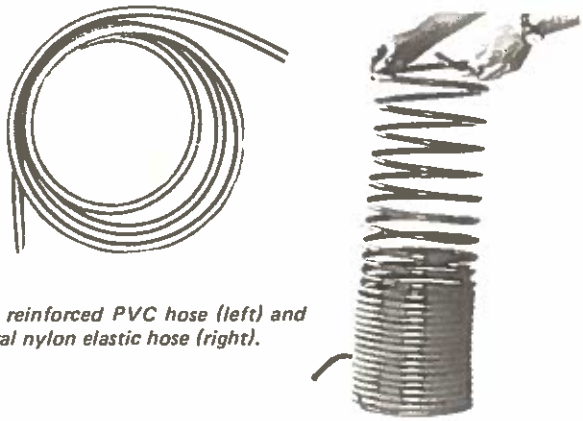
TABLE III – TYPICAL PRESSURE LOSS IN SMOOTH-BORE HOSE (INCLUDING PRESSURE DROP ACROSS STANDARD FITTINGS AT EACH END)

Nominal Bore of Hose BS5118 mm	Gauge Pressure at Hose Inlet bar	Air Flow in litres/second through 15 metres of hose						
		10	16	25	40	60	80	100
		Pressure loss in 'bar' per 15 m length						
12.5	4	0.1	0.3	1.0				
	6	0.06	0.2	0.7	2.1			
19	4	0.02	0.04	0.2	0.4	1.0		
	6	0.01	0.03	0.1	0.2	0.5		
25	4		0.01	0.03	0.1	0.3	0.6	1.0
	6		0.01	0.02	0.05	0.1	0.2	0.3
31.5	4			0.01	0.02	0.07	0.1	0.2
	6				0.01	0.04	0.06	0.07
38	4				0.01	0.02	0.04	0.07
	6					0.01	0.02	0.03

TABLE IV – CHARACTERISTICS OF TYPE SPI NYLON ELASTIC SPIRAL HOSE*

Hose dimensions				Length of hose spiral retracted		Diameter of hose spiral		Maximum working pressure		Working temperature			
bore		outer diameter								maximum		minimum	
mm	in	mm	in	mm	in	mm	in	bar	lb/in²	°C	°F	°C	°F
6.3	1/4	8	5/16	350	13¾	120	4¾	15	210	70	158	–30	–22
10	3/8	12	1/2	400	15¾	160	6.5/16	15	210	70	158	–30	–22
12.5	1/2	16	5/8	420	16½	200	7.7/8	15	210	70	158	–30	–22
20	3/4	25	1	400	15¾	330	13	15	210	70	158	–30	–22

*Atlas Copco



Flexible reinforced PVC hose (left) and spiral nylon elastic hose (right).

TABLE V — EXAMPLES OF SIZES AND RATINGS FOR REINFORCED PVC HOSE

Length		Hose dimensions				Working pressure at +20°C (68°F)		Mass	
		bore		outer diameter					
m	app.ft.	mm	in	mm	in	bar	lb/in^2	kg/m	lb/ft
30	100	3	1/8	8	5/16	20	320	0.07	0.05
30	100	5	3/16	11	7/16	18	260	0.10	0.07
30	100	6.3	1/4	12	15/32	18	260	0.10	0.07
30	100	10	3/8	16	5/8	14	200	0.16	0.11
30	100	12.5	1/2	21.2	27/32	13	185	0.28	0.19
30	100	16	5/8	24.5	31/32	12	170	0.33	0.22
30	100	20	3/4	27.5	1.3/32	12	170	0.38	0.26

form of elastic spiral hose for the connection of air tools, maximum working pressure in this case being 15 bar (215 lb/in^2) (see Table IV).

Heavy duty rubber tubing of reinforced construction is used for hose that has to withstand rough treatment (*eg* on mining and construction sites). Such hose normally consists of a smooth inner tube of rubber reinforced with diagonal cord layers and fitted with a tough, weather-resistant outer cover. On larger sizes (*eg* 80 mm bore and above), as many as six plies may be used in the construction.

Plain flexible PVC tubing has limited application, but in reinforced form can be superior to rubber hose, with superior flexibility which is retained at lower temperatures than other types of plastic hose. Working pressure for PVC hose is normally quoted for room temperature. If used at the maximum service temperature for the material of circa 60°C, working pressure should be reduced by 50% — see also Table V.

See also chapter on *Compressor Air Mains.*

Pressure Drop Calculations

PRESSURE DROP through an air line depends on the flow rate, pipe diameter, pipe length and pipe geometry (*ie* bends, *etc*). It can be determined directly for straight pipes of any given length from empirical formulas, or from tabular or graphical data. Tabular data commonly quote pressure drop per unit length of 100 metres or IS metres (or in original English units, pressure drop per 100 ft length). Pressure drop through valves and fittings, *etc*, is normally quoted in terms of *equivalent pipe length* (*ie* the length of straight pipe of the same diameter size giving the same pressure drop).

Pipe length, however, is relatively insignificant as a design control parameter compared with pipe bore size, since pressure drop is inversely proportional to $(\text{bore})^5$, approximately. In other words, a small change in bore size can have a marked effect on pressure drop, whereas even doubling the pipe length will only result in doubling the pressure drop.

Determination of the relationship between flow rate and pressure drop leads to complex formulas, tedious to manipulate. For most practical purposes a basic formula of the following order can be used

$$\Delta P = \frac{f \cdot L \cdot Q}{r d^x}$$

where

- ΔP = pressure drop
- f = a friction factor
- L = pipe length
- Q = flow rate
- r = compression ratio at pipe entrance
- d = pipe bore
- x = an exponential, the theoretical value of which is 5.33 although 5.0 is more convenient to use
- y = an exponential theoretically equivalent to 1.85 although sometimes taken as 2.0 for simplicity

A practical form of this equation is

$$\Delta P = \frac{f \cdot L \cdot Q^{1.85}}{P_m \cdot d^5}$$

where ΔP = pressure drop in bar
f = friction factor = 500 for conventional steel pipes
L = pipe length in metres
Q = volume rate of flow, litres/sec (free air)
P_m = mean absolute pressure in bar
d = bore of pipe, mm

The nomogram of Fig 1 can be used for obtaining direct solutions, using working pressure in bars.

A limitation of this formula (and the nomogram solutions) is that it does not account for variations in friction factor with pipe bore condition. It will thus over-estimate pressure drop in the case of smooth bore pipes (*eg* copper, nylon and ABS tubes). Since this is on the 'favourable' side, such difference is usually ignored in practice. Rather less significant is that the friction factor will also vary with Reynold's Number (and thus with pipe diameter). Again this can be ignored for most practical calculations involving industrial compressed air supplies.

Pressure Drop Through Bends and Fittings

Pressure drop through bends and fittings can only be determined accurately by empirical tests since it is specific to the internal geometry involved.

Fitting resistance can be expressed in the form of a friction factor or *fitting resistance coefficient,* which can then be used for calculating pressure drop in the same formula as for pipes. Typical resistance coefficients are given in Table I. In the case of proprietary fittings, this will only give accurate results if the specific value of the resistance coefficient has been determined at the anticipated Reynold's Number of flow, *ie* representative flow velocity for a given fitting size.

It is generally more convenient and simpler to calculate fitting resistance in terms of equivalent length of straight pipe — *ie* the length of straight pipe, of the same bore size as the fitting, which will have the same resistance (and thus pressure drop) as the fitting. Again such values can only be determined on empirical lines (see Tables II, III and IV), but the following formula will be found to be in close agreement with the results obtained in practice:-

$$L_f = KfD^{1.2}$$

where L_f = equivalent length of straight pipe
K = is a constant = 4.34 for L in metres and D in centimetres
= 43.7 for L in feet and D in inches
f = fitting resistance coefficient
D = pipe or fitting bore size

The accuracy of this formula depends on the accuracy of the frictional coefficient used. For non-critical applications tabular data are usually referred to.

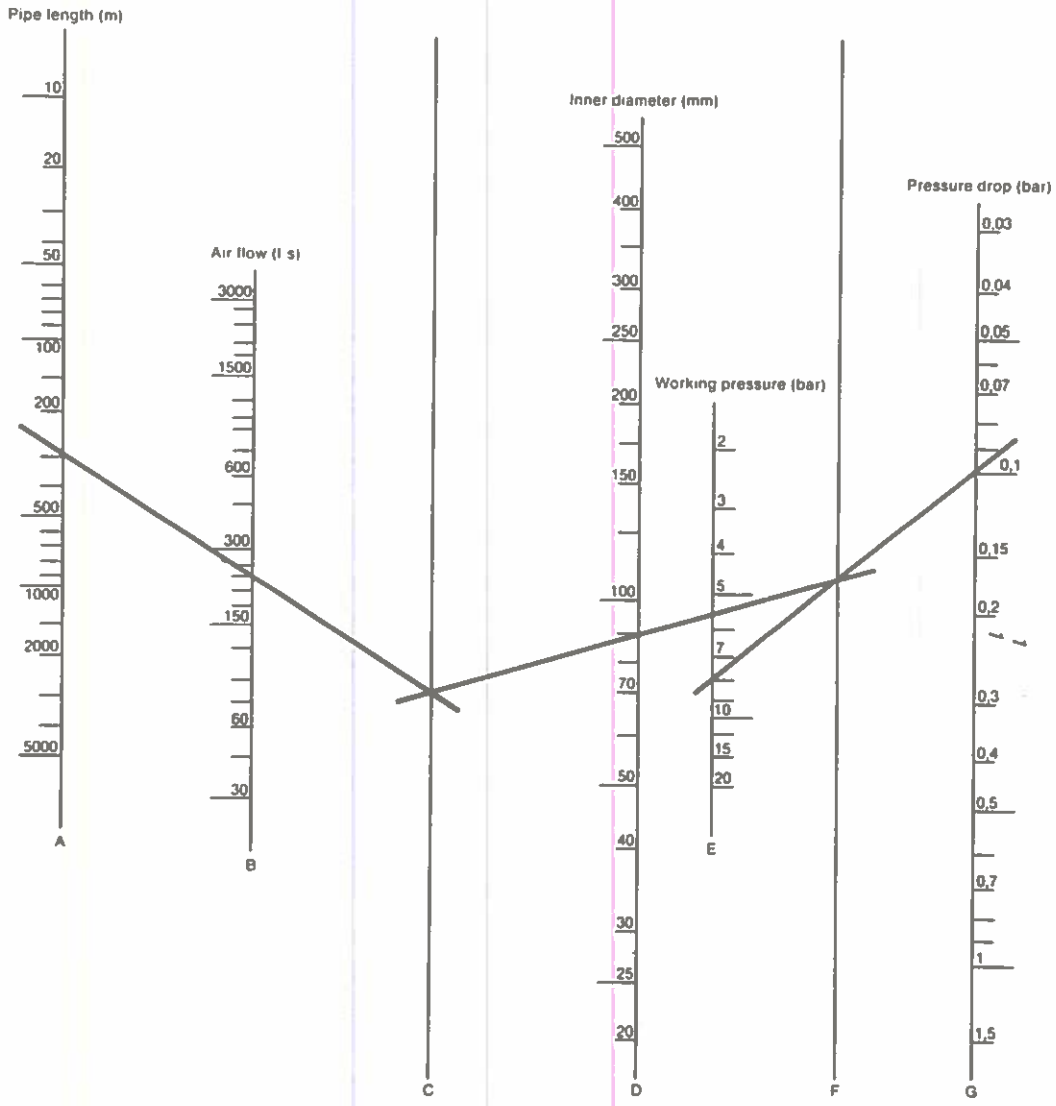


Fig 1 Pressure drop nomogram.
Connect the pipe length (A) with the air flow (B) and extend to line C. The intersection on line C is then connected with the pipe diameter (D) and extended to line F. With the intersection on line F as pivot connect the actual working pressure (E) with scale G and read the pressure drop.
(Atlas Copco).

TABLE I – RESISTANCE COEFFICIENTS OF FITTINGS AND BENDS

Fitting	Order of Resistance Coefficient
Large elbow	0.33
Small elbow	0.42
90° sharp elbow	0.50–1.00
Large tee, straight through	0.33
Large tee, branch flow	1.00–1.33
Small tee, straight through	0.40–0.50
Small tee, branch flow	1.30–1.50
Cross, straight through	0.35–0.50
Cross, branch flow	1.00–1.40
Close return bend	0.60–0.90

TABLE II – PRESSURE LOSS THROUGH STEEL FITTINGS – EQUIVALENT PIPE LENGTHS

Item	Equivalent pipe length in metres									
	Inner pipe diameter in mm									
	15	20	25	40	50	80	100	125	150	200
Gate valve fully open	0.1	0.2	0.3	0.5	0.6	1.0	1.3	1.6	1.9	2.6
half closed		3.2	5	8	10	16	20	25	30	40
Diaphragm valve fully open	0.6	1.0	1.5	2.5	3.0	4.5	6	8	10	
Angle valve fully open	1.5	2.6	4	6	7	12	13	18	22	30
Globe valve fully open	2.7	4.8	7.5	12	15	24	30	38	45	60
Swing check valve fully open		1.3	2.0	3.2	4.0	6.4	8.0	10	12	16
Bend R = 2d	0.1	0.2	0.3	0.5	0.6	1.0	1.2	1.5	1.8	2.4
Bend R = d	0.2	0.3	0.4	0.6	0.8	1.3	1.6	2.0	2.4	3.2
Mitre bend 90°	0.6	1.0	1.5	2.4	3.0	4.8	6.0	7.5	9	12
Run of tee	0.2	0.3	0.5	0.8	1.0	1.6	2.0	2.5	3	4
Side outlet tee	0.6	1.0	1.5	2.4	3.0	4.8	6.0	7.5	9	12
Reducer		0.3	0.5	0.7	1.0	2.0	2.5	3.1	3.6	4.8

TABLE III – PRESSURE LOSS THROUGH ABS FITTINGS – EQUIVALENT PIPE LENGTHS

Pipe outside diameter	equivalent pipe length in metres							
	16	25	32	50	63	75	90	110
90° elbow	0.34	0.5	0.65	1.0	1.26	1.5	1.88	2.58
45° elbow	0.16	0.24	0.32	0.52	0.63	0.75	0.95	1.33
90° bend	0.1	0.16	0.22	0.34	0.44	0.56	0.75	1.00
Tee in line flow	0.123	0.19	0.23	0.36	0.45	0.56	0.69	0.95
Tee in line to branch flow	0.77	1.17	1.47	2.21	2.98	3.68	4.57	6.00
Reducer	0.22	0.31	0.37	0.51	0.80	1.11	1.34	1.58

TABLE IV – EQUIVALENT PIPE LENGTHS OF VALVES AND BENDS*

Item	Equivalent pipe length in m										
	Inner pipe diameter in mm										
	25	40	50	80	100	125	150	200	250	300	400
gate valve, fully open half closed	0.3 5	0.5 8	0.6 10	1.0 16	1.3 20	1.6 25	1.9 30	2.6 40	3.2 50	3.9 60	5.2 80
diaphragm valve fully open	1.5	2.5	3.0	4.5	6	8	10	–	–	–	–
angle valve, fully open	4	6	7	12	15	18	22	30	36	–	–
globe valve, fully open	7.5	12	15	24	30	38	45	60	–	–	–
swing check valve fully open	2.0	3.2	4.0	6.4	8.0	10	12	16	20	24	32
bend R=2d	0.3	0.5	0.6	1.0	1.2	1.5	1.8	2.4	3.0	3.6	4.8
bend R=d	0.4	0.6	0.8	1.3	1.6	2.0	2.4	3.2	4.0	4.8	6.4
mitre bend, 90	1.5	2.4	3.0	4.8	6.0	7.5	9	12	15	18	24
run of tee	0.5	0.8	1.0	1.6	2.0	2.5	3	4	5	6	8
side outlet tee	1.5	2.4	3.0	4.8	6.0	7.5	9	12	15	18	24
reducer	0.5	0.7	1.0	2.0	2.5	3.1	3.6	4.8	6.0	7.2	9.6

*Atlas Copco

Pressure Drop Through Flexible Pipes

Theoretically, at least, if a flexible pipe has a smooth bore and a straight run, the pressure drop will be the same as that of a rigid smooth bore pipe of the same bore size. Flexible pipes are, however, liable to manipulation through various forms of bends, with consequent increased and variable resistance to flow; and the flow itself may be of a fluctuating or pulsating nature. Line sizing in this case is thus normally based on practical recommendations – see Table V. Likely losses can be estimated from the tabular data. Only in critical systems is it likely to be necessary to determine the actual pressure drop produced by specific tests on a sample length of hose manipulated under design working conditions.

TABLE V – APPROXIMATE PRESSURE DROP IN FLEXIBLE HOSE WITH COUPLINGS EACH END

(pressure drop in bar per 30 metres of hose length)

Hose Bore	Gauge Pressure bar	FREE AIR FLOW litres/sec													
		10	15	20	25	30	35	40	45	50	55	60	65	70	75
12.5 mm (½ inch)	3.5	0.28	0.7	1.40	2.5	–	–	–	–	–	–	–	–	–	–
	4	0.22	0.6	1.30	2.0	3.25	–	–	–	–	–	–	–	–	–
	5	0.18	0.5	1.20	1.8	2.75	–	–	–	–	–	–	–	–	–
	5.5	0.16	0.45	1.10	1.6	2.40	3.45	4.75	–	–	–	–	–	–	–
	6	0.14	0.4	1.00	1.4	2.10	3.00	4.20	5.70	–	–	–	–	–	–
	7	0.10	0.35	0.08	1.2	1.85	2.70	3.70	5.00	–	–	–	–	–	–
	7.5	0.07	0.3	0.6	1.0	1.65	2.40	3.40	4.60	6.10	–	–	–	–	–
19 mm (¾ inch)	3.5	0.07	0.15	0.20	0.35	0.50	0.55	0.90	1.17	1.60	1.95	–	–	–	–
	4	0.06	0.13	0.18	0.32	0.45	0.50	0.85	0.95	1.40	1.75	–	–	–	–
	5	0.05	0.11	0.15	0.30	0.40	0.50	0.80	0.80	1.20	1.55	1.50	–	–	–
	5.5	0.04	0.10	0.13	0.27	0.35	0.45	0.70	0.75	1.10	1.30	1.20	1.50	–	–
	6	0.03	0.08	0.11	0.25	0.30	0.40	0.60	0.70	1.00	1.10	1.00	1.30	–	–
	7	0.02	0.06	0.09	0.20	0.75	0.35	0.50	0.60	0.80	0.90	0.90	1.15	1.30	1.50
	7.5	0.01	0.04	0.07	0.15	0.20	0.30	0.40	0.50	0.60	0.70	0.85	1.00	1.20	1.40
25 mm (1 inch)	3.5	0.01	0.04	0.07	0.09	0.15	0.16	0.20	0.30	0.35	0.50	0.70	0.10	–	–
	4	0.01	0.03	0.07	0.08	0.13	0.15	0.18	0.25	0.30	0.40	0.50	0.70	0.85	1.00
	5	–	0.03	0.06	0.07	0.11	0.13	0.15	0.20	0.25	0.37	0.45	0.60	0.70	0.80
	5.5	–	0.02	0.05	0.06	0.09	0.12	0.13	0.18	0.22	0.33	0.42	0.50	0.55	0.60
	6	–	0.02	0.04	0.05	0.08	0.11	0.12	0.16	0.20	0.27	0.32	0.40	0.45	0.50
	7	–	0.01	0.03	0.04	0.07	0.09	0.11	0.14	0.18	0.27	0.26	0.30	0.35	0.40
	7.5	–	0.01	0.03	0.03	0.05	0.07	0.10	0.13	0.16	0.19	0.27	0.25	0.27	0.30
30 mm (1¼ inch)	3.5	–	–	0.03	0.03	0.04	0.06	0.08	0.10	0.12	0.15	–	–	–	–
	4	–	–	0.02	0.02	0.03	0.06	0.07	0.08	0.10	0.12	0.15	0.15	0.20	–
	5	–	–	0.01	0.02	0.03	0.05	0.06	0.07	0.08	0.10	0.13	0.12	0.18	0.20
	5.5	–	–	–	0.01	0.02	0.04	0.05	0.06	0.07	0.09	0.12	0.10	0.16	0.20
	6	–	–	–	–	0.02	0.03	0.04	0.05	0.06	0.08	0.10	0.09	0.12	0.15
	7	–	–	–	–	–	0.02	0.03	0.04	0.05	0.07	0.08	0.08	0.10	0.10
	7.5	–	–	–	–	–	–	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.10
38 mm (1½ inch)	3.5	–	–	–	–	–	0.02	0.03	0.04	0.05	0.05	0.06	0.06	0.07	0.07
	4	–	–	–	–	–	–	0.02	0.03	0.04	0.04	0.05	0.06	0.06	0.07
	5	–	–	–	–	–	–	–	0.02	0.03	0.03	0.04	0.05	0.06	0.07
	5.5	–	–	–	–	–	–	–	–	0.02	0.02	0.03	0.04	0.05	0.06
	6	–	–	–	–	–	–	–	–	–	0.02	0.02	0.03	0.04	0.05
	7	–	–	–	–	–	–	–	–	–	–	0.02	0.02	0.03	0.04
	7.5	–	–	–	–	–	–	–	–	–	–	–	0.02	0.03	0.035

Losses in End Fittings

Losses developed in end fittings can have a significant effect on the overall pressure drop, as in addition to the free flow area an *orifice loss* is also involved. This latter figure can be expressed in terms of an *orifice coefficient,* the value of which will always be less than unity (and will thus represent a relative loss or throttling effect).

Where the orifice coefficient for the end fitting is known, then the airflow through the fitting can be calculated from the following simple formula, which is sufficiently accurate for general use:

$$\text{air flow} = A \cdot c \cdot d\sqrt{P}$$

where

- c = orifice coefficient
- d = orifice diameter
- P = air pressure at the entry to the orifice (*ie* taking into account the total pressure drop to this point).

A is a constant which has the following specific values:

- A = 2.55 for c and d in millimetres and pressure in bars
- = 115 for c and d in inches and pressure in lb/in^2

Blowers

Where comparatively large flow rates are involved with low initial pressures (*eg* blower applications), the following simplified formula can be used:-

$$\Delta P = \frac{F}{P_m} \times \frac{L}{1000}$$

where

- P_m = mean pressure (absolute)
- F = a friction factor

$$= \frac{kV^{1.85}}{d^5}$$

where k is a coefficient depending on the units employed

- k = 482 for V in metres/sec, d in centimetres, ΔP and P_m in bar
- k = 7.0 for V in ft/sec, d in inches, ΔP and P_m in lb/in^2

Pipe Materials

STEEL IS the most common material used for main line pipes, although ABS (thermoplastic polymer) has become an acceptable alternative. Other materials used on smaller mains are:

copper (nominal diameter up to 40 mm)

thin walled stainless steel (nominal diameter up to 25 mm)

rubber hose and plastic tubing

Steel Pipes

Steel pipes to BS1387 (or equivalent) are normally specified in nominal bore sizes (see Table I). These sizes are suitable for screwing to BS21 pipe threads. Preferred finishes are black or galvanized, the latter having better corrosion resistance.

Steel pipes are normally used with threaded connections, but above 65 mm nominal bore welded fittings to BS1965 are recommended. Malleable iron fittings, where used, should conform to BS143 and BS1256.

TABLE I — NOMINAL BORE SIZE STEEL TUBES TO BS1387

Nominal Size	Actual Bore				Approximate o.d.	
	Medium		Heavy			
in	in	mm	in	mm	in	mm
1/8	0.226	5.77	0.178	4.52	13/32	10.3
1/4	0.338	8.59	0.290	7.37	17/32	13.5
3/8	0.476	12.09	0.428	10.87	11/16	17.5
1/2	0.623	15.82	0.575	14.60	27/32	21.5
3/4	0.839	21.31	0.791	20.09	1.1/16	27
1	1.060	26.92	0.996	25.30	1.11/32	34
1.1/4	1.401	35.59	1.337	33.96	1.11/16	43
1.1/2	1.633	41.48	1.569	39.85	1.29/32	48
2	2.066	52.48	2.002	50.80	2.3/8	60

Plastic Pipes

ABS is a tough, ductile and non-corroding polymer and the only plastic material generally recognized as suitable for air mains. Its main limitation is that it softens and loses strength with increasing temperature. Rated pressure is quoted at 20°C with, typically, some 20% derating at 35°C, *eg*

rating at 20°C	rating at 35°C
10 bar	8 bar
12.5 bar	10 bar

The other limitation is that ABS is not suitable for threading so connection to metal components must be made with composite unions or adaptors.

Standard types of ABS pipes produced for air mains are rated in two pressure classes: NP10 (maximum continuous working pressure 10 bar) and NP12.5 (maximum continuous working pressure 12.5 bar).

No other plastic material is (as yet) normally used for compressed air mains, particularly as materials such as rigid PVC can be subject to brittle failure. Flexible plastic tubes are, however, used for branch lines.

Copper Pipes

Copper tubing used for air mains should conform to BS2871 (Part 1), although BS659 and BS2017 sizes remain in use, (see Table II). It is (almost) invariably used with compression fittings which makes it a more expensive alternative to steel or ABS for air mains. Copper piping is, therefore, more widely used for branch lines.

Branch Lines

In addition to copper, nylon tubing is commonly employed for small diameter branch lines, with flexible lines at tool take-off points. Preferred sizes of nylon tubes are: (see also Table III).

o.d. (ins)	i.d. (ins)	o.d. (mm)	i.d. (mm)	BSP Fitting Size
1/8	0.058	5	3.3	1/8
3/16	0.122	8	5.5	1/4
1/4	0.165	12	8.5	3/8
5/16	0.213	15	12.0	1/2
3/8	0.245			
1/2	0.369			
3/4	0.587			

Flexible nylon tubes may be used directly for small diameter hose or reinforced with braid for larger hose sizes. Coiled nylon tubes are also used for flexible lines, the tubing in this case being of small bore (*eg* up to 5/8 inch) formed in the shape of a close-wound coil. This results in an extremely flexible line which can also be extended considerably in length, retracting and re-stacking to its original coil form when released.

Another flexible form of air line hose is PVC reinforced with terylene. Working pressure rating is similar to that of plain nylon tubing, *eg* 17.5–12 bar (250–170 lb/in^2), depending on size. Bore sizes up to about 25 mm (1 inch) are readily obtainable. Working temperature range is, however, limited to about −10°C to +40°C (14°F to 104°F). Reinforcement may be mesh type (*ie* yielding

TABLE II – COPPER PIPE SIZES FOR AIR LINES

o.d. in BS2017	Bore in	o.d. mm	Equivalent in		o.d. BS659	Bore in	o.d. mm BS2871	Bore mm
1/8	0.075	–	–		3/16	1/8	–	–
3/16	0.131	5	13/64		–	–	5	3.4
1/4	0.178	8	5/16		11/32	1/4	8	6.4
3/8	0.279	–	–		15/32	3/8	–	–
1/2	0.410	12	15/32		19/32	1/2	12	10.4
3/4	0.655	16	5/8		27/32	3/4	15	13
1	0.872	22	7/8		–	–	–	–
1.1/4	1.122	28	1.1/8		–	–	–	–

TABLE III – PROPERTIES OF NYLON TUBING

o.d. in	i.d. in	Burst Pressure at 20°C lb/in^2	bar	Min Bend Rad	Weight (lbs) per ft
Light Gauge					
1/8	0.095	1 000	67	3/4	0.0023
3/16	0.137	1 000	67	1.1/4	0.0060
1/4	0.190	1 000	67	2	0.0100
5/16	0.242	1 000	67	2.1/4	0.0140
3/8	0.295	1 000	67	3.1/2	0.0200
1/2	0.406	900	60	4.1/2	0.0370
5/8	0.500	800	54	6	0.0500
Heavy Gauge					
1/8	0.075	2 250	150	3/4	0.0140
3/16	0.107	2 250	150	1.1/4	0.0033
1/4	0.150	2 250	150	1.3/4	0.0090
5/16	0.182	2 250	150	2.1/4	0.0230
3/8	0.225	2 250	150	3	0.0310
1/2	0.375	2 000	133	4	0.0600

a transparent tube) or an opaque braid. Many of the earlier limitations of plastic hoses, *eg* particularly lack of flexibility at lower temperatures, have been overcome with later formulations. Spiral wire reinforced plastic tubes may be used for hose duties, the latest types offering a smooth rather than a corrugated bore. Spiral wire reinforced plastic tubes resist deformation under negative (suction) as well as positive internal pressures.

Rubber hose is generally used for flexible lines where a wider working temperature is required or larger hose sizes are needed. Air line hoses are normally manufactured to BS924 Type A or B requirements (or American or Continental equivalents). The basic construction comprises a seamless inner tube of oil-resistance rubber, wrapped with plies of cotton and a smooth outer cover.

Typical sizes and ratings are:-

Type A — sizes 4.8–9.5 mm (3/16 in to 3/8 in) — bore pressure rating 12 bar (170 lb/in^2) maximum

Type B — 12.5–63.5 mm (½ in to 2½ in) bore.

Typical pressure ratings:-

mm	in	bar	lb/in^2
4.8	3/16	32	450
6.3	1/4	31	440
8.0	5/16	26	375
9.5	3/8	22	310
12.5	1/2	19	275
19.0	3/4	16	225
25.0	1	14	200

Rubber hoses and flexible lines for suction services normally feature wrapped ply construction. The fabric reinforcement is laid in the form of several close layers or plies, rather than braid, producing a more rigid construction. Where greater flexibility is required, the plies may be separated or cushioned. This also improves the kink-resistance of the hose. Metallic wire reinforcement may also be used in the form of a wire spiral, embedded or semi-embedded in the liner. Only fully embedded spirals produce a smooth bore tube.

Plain (unreinforced) polythene tubing is more flexible than nylon tubing. Typical maximum pressure rating is 7 bar (100 lb/in^2) for 15 mm (5/8 in) o.d. tubing and up to 14 bar (200 lb/in^2) for 5 mm (3/16 in) o.d. tubing. Flexible polythene tubing is not suitable for pressure use at temperatures much above 75°C, nor for rigid installations except where adequate support by pipe clips can be arranged.

Flexible PVC tubing is only suitable for low pressure and has a maximum service temperature rating of 70°C. Its flexibility can compare with, or is better than, that of polythene tubing.

Tube and Pipe Bending

Copper is an attractive material for rigid pipes since it is readily manipulated into bends, enabling the number of fittings required to be reduced. It can show considerable cost advantages over steel or ABS in this respect.

Small diameter metallic pipes and tubes can be manipulated by hand for bending, but care is needed to ensure that the tube is not kinked or otherwise deformed, which will restrict the air flow. The absolute minimum for the radius of bend is three times the tube o.d., and a higher figure should always be adopted for hand bending. Larger radius bends will also have lower frictional losses. Fillers may be used to support the inside of pipes and tubes for hand bending. These are effective only if the tube is filled 'solid'. Low melting point metallic fillers are generally to be preferred, and are readily removed by gentle heat after bending. The material is re-usable. Sand is not recommended as a filler due to the difficulty of ensuring its complete removal unless pressure cleaning equipment is available.

Proprietary pipe benders are available for manipulating larger sizes of pipes and tubes and also for handling all thin walled metallic tubing, which is difficult to bend without local deformation.

These work on one of the following principles and can produce satisfactory cold bends.

Compression bending — where the tube is clamped and effectively put in compression during bending. It is most suitable for thicker walled tubing and larger bend radii. Its main advantage is that the construction of the bender is simple and the bending operation can be completed rapidly.

Draw bending — where the tubing is clamped to a grooved form and literally wound to the bend by the action of a moving shoe. It is suitable for the tightest bends, but there may be some deformation of the section through flattening.

Press bending — a simple and rapid technique, where the tube is manipulated by rollers working against a reaction block. It is mostly suited to thicker walled tubing and generous bend radii.

Roll bending — particularly suited to the manipulation of thin walled tubes, involving the passing of the tube through a series of offset grooved rollers. Some of the rollers are power driven, the others are idlers. The driven rollers are brought progressively nearer to each other, until the required radius of bend is obtained. Such machines can also produce complete 360° bends and coils.

Pipe Supports

Permanent installations in rigid (metallic) tubing must always be adequately supported so that no strain is put on the compressor or other devices to which they are attached. Supporting clips should, preferably, be of the type with elastomeric liners or similar feature, which clamp the pipe or tube rigidly, but at the same time allow for axial movement to accommodate contraction or expansion. Alternatively, for shorter runs, thermal expansion and contraction can be accommodated by bends or, in the case of large pipes, the fitting of expansion joints.

Clip spacing is generally less critical in the case of nylon tubing, although again the question of avoiding snagging is important. It may also be desirable to anchor the tubing securely on either side of bends to prevent whipping or excessive displacement which could follow an interruption of flow.

Clips should be used on pipes close up to fittings, *etc*, which are not themselves mounted to save the pipe from carrying the weight of the fitting. Logically, too, the complete line system should be laid out as neatly and tidily as possible, although this consideration is often modified by the respective lengths of lines involved. However, the shortest distance between two points is not necessarily the best run for a line. Conversely, designing a 'tidy' system by using a multiplicity of elbows and fabricated bends to group runs of lines, may represent unnecessary cost and unnecessary additional pressure drop.

In certain systems it may be desirable that all piping be electrically bonded. This can be done by using pipe clips which incorporate a small tongue or similar projection in contact with the surface of the pipe, when the pipe may be earthed through the clip. With plain clips, a bond can be formed by inserting a layer of metallic gauze to a suitable point.

Simple pipe clips are commonly manufactured in brass or copper. These may be shaped to hold individual pipes of a specified overall diameter, or up to six pipes in a single clip. Unless insulated from the pipe, *eg* with a plastic liner, pipe clips should always be of similar metal to the pipes to avoid the possibility of electrolytic corrosion.

See also chapters on *Compressed Air Mains* and *Branch Lines*.

Compressed Air Treatment

TYPES AND sources of contaminants in compressed air systems and methods of treatment are summarized in Table I. The degree of treatment necessary depends on the particular application. Thus, for general industrial applications using air mains for compressed air supply, intake filtering at the compressor and aftercooling of the air delivered by the compressor to provide partial water removal, and finally followed by filters to receive solid contaminants down to a specific size may be adequate. In other systems two or more stages of filtration/separation may be required; and in more critical applications this may be further followed by drying to provide humidity control.

TABLE I

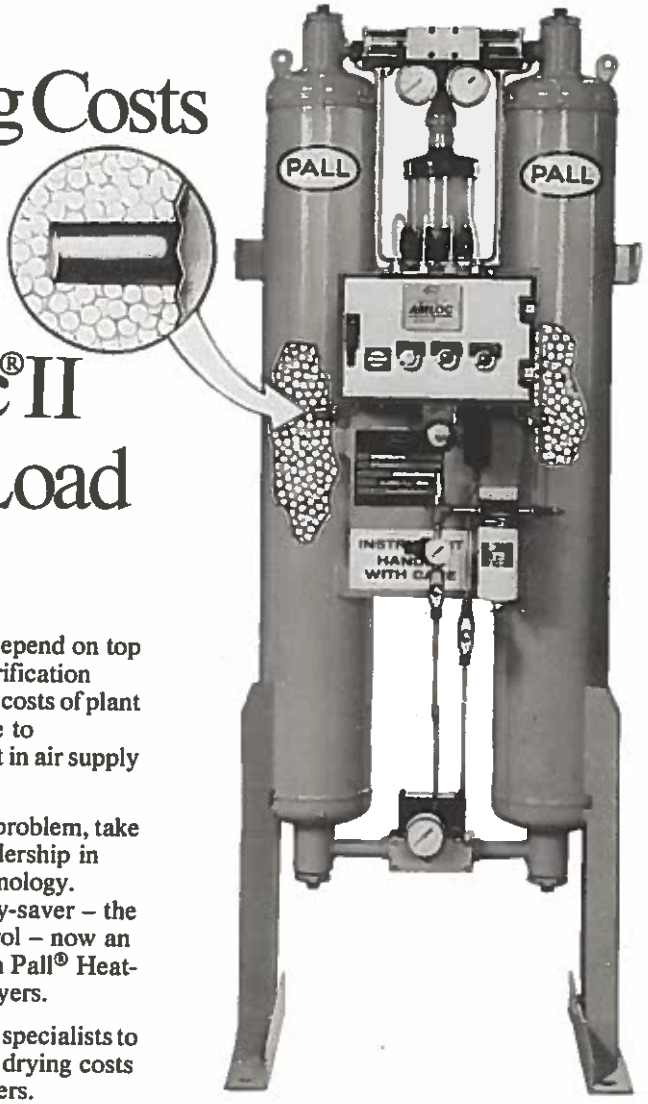
Contaminant	Source	Entry to System	Removal
dust, smoke, fumes	atmosphere	inducted by compressor	intake filter
bacteria, viruses	atmosphere	inducted by compressor	sterilizer
water vapour	atmosphere	inducted by compressor	separator followed by dryer if necessary
gases	compressor	generated in compressor	
oil	compressor	carried over from compressor	separator
solids	system corrosion products	from walls of air mains etc	line filters

Intake Filters

Large stationary compressors are commonly installed in separate rooms drawing air from the outside atmosphere. This has the advantage of eliminating induction of factory-generated fumes, contaminants, *etc*. The level of dust concentration present in the inducted air will obviously vary with different sites, but can be expected to be of the order of 10–50 mg/m^3 (0.6–3.0 x 10^{-6} lb/ft^3).

Standard practice is to fit the compressor with an intake filter having an efficiency of the order of 99.9% – *eg* cartridge type filter with filter element, or panel filter, depending on the size of the compressor. Such filters can be expected to pass all solid particles smaller than 5 μm in addition to

Cut Drying Costs even more with the Pall® Amloc® II Moisture-Load Control



Companies all over the world depend on top quality Pall compressed air purification systems to reduce the expensive costs of plant and equipment breakdowns due to contaminating water, oil and dirt in air supply systems.

Whatever your compressed air problem, take advantage of our 30 years' leadership in filtration and super drying technology. Benefit from Pall's latest energy-saver – the Amloc II Moisture-Load Control – now an exclusive built-in feature of both Pall® Heat-Les® and Heat Reactivated Dryers.

Write or telephone for one of our specialists to advise you on how to cut **your** drying costs with Pall high performance dryers.

® *Pall, Amloc and Heat-Les are Registered Trade Marks.*

Pall® Heat-Les® Dryer

PALL

PALL PNEUMATICS LIMITED
A subsidiary of Pall Europe Limited
Europa House, Havant Street, Portsmouth PO1 3PD. England.
Tel: (0705) 753545 Telex: 86251 Cables: Pall Portsmouth Telex
With international Sales Offices, Plants and Distributor Services throughout the world.

PSIΨ THE FILTER SYSTEM

A full range of Filters and Silencers to meet your requirements

Specialists in Ultra High Efficiency Filters, for the removal of oil from compressed air applications

The Filter Silencer – a new innovation by PSI.Ψ
A product specifically designed to reduce noise levels and regenerate exhausting compressed air

PROCESS SCIENTIFIC INNOVATIONS LIMITED
INDUSTRIAL ESTATE, BOWBURN
DURHAM, ENGLAND DH6 5AD
Tel: (0385) 770550 Telex: 537383

PSIΨ

atmospheric water vapour, gases, vapours, odours, bacteria and rinses which may also be present in the inducted air. Cut-off will not be positive at 5 μm and a typical 'count' of larger solids passed would be:

5 μm	–	12% of total weight
5 – 10 μm	–	12% of total weight
10 – 20 μm	–	14% of total weight
20 – 40 μm	–	23% of total weight
40 – 80 μm	–	30% of total weight
80 – 200 μm	–	9% of total weight

Intake filters may be of the 'dry' or 'wet' type. Compressor manufacturers will normally specify which type and size of filter to use for specific applications, or supply suitable general purpose filters as standard fittings. The intake filter may also be combined with a silencing chamber (filter silencer), or a separate air silencer fitted after the filter to suppress a compressor noise. Where air is inducted from an outside source the filter would be fitted at the point of air entry to the building rather than on the compressor itself. In this case it is important that the pipework between the filter and compressor be clean and free from corrosion. If any doubts exist on this score, an additional filter or simple filter screen can be fitted at the compressor inlet.

Types of Intake Filters (see also Fig 1)

Types of intake filters most commonly used on compressors are:

(i) *Paper filters* – with renewable elements, used on all types of compressors. Filtering efficiency is high (*eg* over 99%) with typical pressure drop for a new element

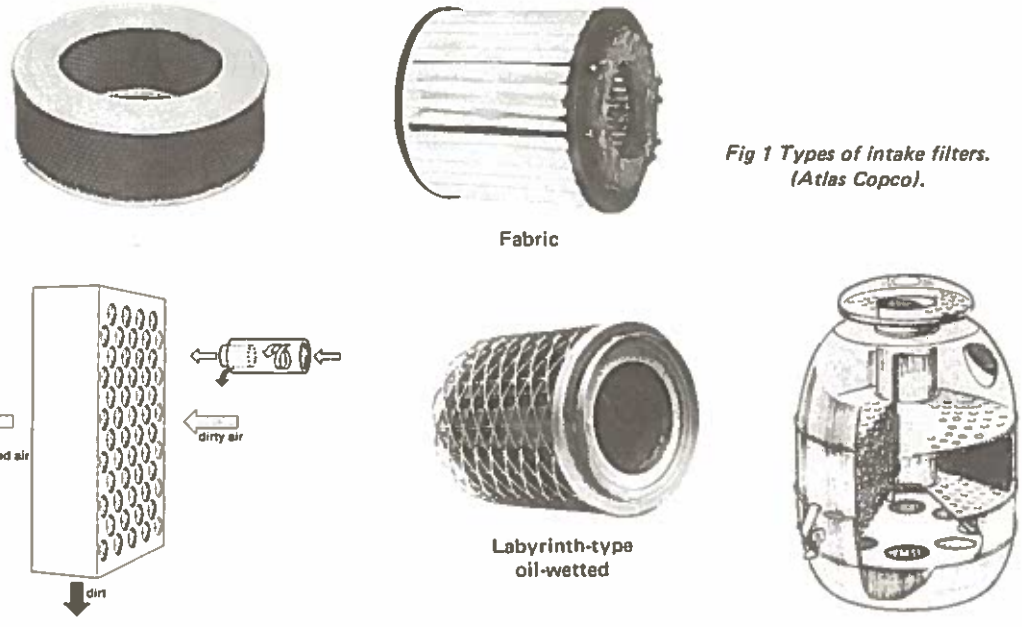


Fig 1 Types of intake filters. (Atlas Copco).

of 2.5–3.5 millibar if correctly sized. Paper filters are not generally recommended for reciprocating compressors unless a pulsation chamber is incorporated between the filter and compressor intake. They are also not suitable for handling air at temperatures in excess of 80°C.

(ii) *Fabric filters* – which are stronger than paper filters and generally cleanable by back-flow of compressed air, or in some cases by washing.

(iii) *Oil-bath filters* – which have a higher solids retention than the aforementioned types with a capacity of collecting impurities equal to the mass of the oil. They are not suitable for use on piston compressors unloaded by closing the air inlet valve unless installed with a bypass or check valve to prevent oil being blown back out of the filter with reverse air flow.

(iv) *Labyrinth type oil-wetted filters* – mostly used on small compressors inducting relatively clear air. They require periodic cleaning.

(v) *Inertial filters* – of reverse flow or cyclonic type suitable for coarse filtering of large volumes of air. They are normally used as prefilters in combination with paper, fabric or oil-bath filters.

The most suitable type of intake filter is normally specified by the compressor manufacturer, based on the degree of filtration desired and the operating conditions.

Water Separation

Ambient air always contains water vapour. After leaving the compressor the air is fully saturated with water vapour, the actual amount of water vapour present being directly proportional to the temperature of the air and inversely proportional to the pressure (see Tables IIA and IIB). Thus liquid water is best removed when the temperature of the air is lowest and the pressure is at its highest – *ie* immediately on leaving the compressor.

Standard practice is to follow the compressor with an aftercooler which should be of sufficient size and cooling capacity to reduce the temperature of the outgoing air to within 8–10 degC of the temperature of the water entering the aftercooler. Cooling in the aftercooler may be by ambient air or water. In the latter case approximately 20 litres of water will be required for every 2.5 m^3 of free air being cooled in a typical system to realize the aforementioned performance.

A receiver following the aftercooler, and located in the coolest place possible, will permit further cooling and water condensation – Fig 2.

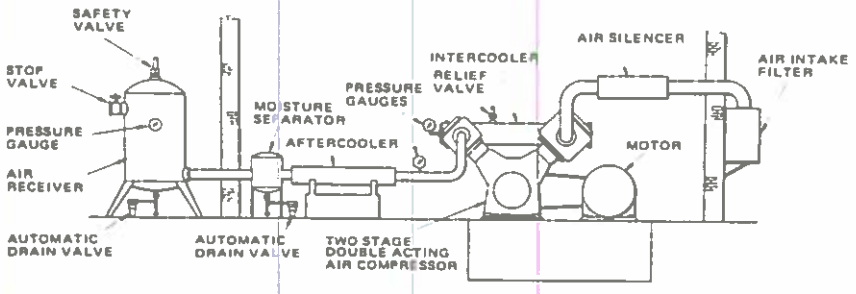


Fig 2 A typical compressor installation. (IMI Norgren)

TABLE IIA – MOISTURE CONTENT OF SATURATED COMPRESSED AIR

Example – 1 m^3 of air at atmospheric conditions fully saturated at 20°C contains 17.4 g of water vapour.
When compressed to 6.3 bar and 50°C it can only retain 11.2 g as vapour ∴ 6.2 g (17.4–11.2) is released as liquid water.
If cooled down to 25°C a further quantity of 7.93 g (11.2–3.27) of water will condense out

Air Temp °C	WATER VAPOUR CONTENT OF FULLY SATURATED COMPRESSED AIR Grams of water vapour per cubic metre (g/m^3) of air at standard atmospheric pressure of 1013 mbar (0 bar gauge) and saturation conditions and when compressed to the pressure and temperature shown												
	AIR PRESSURE IN BARS												
	0	0.4	0.63	1.0	1.6	2.5	4.0	6.3	8.0	10.0	12.5	16.0	20.0
0	4.82	3.45	2.97	2.42	1.87	1.39	0.97	0.67	0.54	0.44	0.36	0.29	0.23
5	6.88	4.93	4.24	3.46	2.68	1.99	1.39	0.95	0.77	0.63	0.52	0.41	0.33
10	9.41	6.74	5.80	4.73	3.66	2.72	1.90	1.30	1.06	0.87	0.70	0.56	0.45
15	12.7	9.08	7.83	6.39	4.94	3.67	2.56	1.76	1.43	1.17	0.95	0.76	0.61
20	17.4	12.5	10.7	8.75	6.77	5.02	3.51	2.41	1.95	1.60	1.30	1.04	0.84
25	23.6	16.9	14.6	11.9	9.18	6.82	4.77	3.27	2.65	2.17	1.77	1.40	1.14
30	30.5	21.8	18.8	15.3	11.9	8.81	6.16	4.22	3.43	2.81	2.29	1.81	1.47
35	39.0	27.9	24.0	19.6	15.2	11.3	7.87	5.40	4.38	3.59	2.92	2.32	1.88
40	49.6	35.5	30.6	24.9	19.3	14.3	10.0	6.87	5.57	4.55	3.72	2.95	2.39
45	63.5	45.5	39.2	31.9	24.7	18.3	12.8	8.79	7.13	5.84	4.76	3.77	3.06
50	81.0	58.0	49.9	40.7	31.5	23.4	16.4	11.2	9.10	7.45	6.07	4.82	3.90

TABLE IIB – MOISTURE CONTENT OF SATURATED COMPRESSED AIR

Air Temp °F	Air Pressure – Pounds per Square Inch (lb/in^2) gauge												
	0	30	40	50	60	70	80	90	100	110	120	130	150
32	4.34	1.43	1.16	0.98	0.85	0.75	0.67	0.61	0.55	0.51	0.47	0.44	0.38
40	6.00	2.09	1.60	1.36	1.16	1.03	0.91	0.84	0.77	0.70	0.65	0.60	0.52
50	8.69	2.87	2.35	1.93	1.70	1.52	1.43	1.23	1.12	1.02	0.95	0.89	0.79
60	12.41	4.10	3.32	2.80	2.35	2.13	1.90	1.77	1.58	1.46	1.35	1.26	1.10
70	17.85	5.90	5.05	4.08	3.52	3.10	2.66	2.52	2.28	2.10	1.95	1.82	1.59
80	24.97	8.25	6.80	5.70	4.87	4.35	3.88	3.52	3.20	2.93	2.78	2.53	2.23
90	30.81	10.20	9.33	7.90	6.78	6.00	5.33	4.90	4.42	4.06	3.76	3.50	3.13
100	43.40	14.30	12.70	10.80	9.28	8.25	7.32	6.93	6.03	5.57	5.15	4.80	4.20
110	64.50	21.30	18.80	14.60	12.60	11.10	9.85	8.98	8.12	7.46	6.93	6.45	5.66
120	87.00	28.70	23.20	19.70	16.80	14.80	13.30	11.60	10.90	10.10	9.30	8.67	7.56

For compressors operating at pressures in the region of 7 bar gauge the size of the receiver in litres should be approximately equal to thirty times the rated free air delivery of the compressor in dm^3/s, *ie* a compressor rated at 50 dm^3/s free air delivery requires a receiver of approximately 1500 litres capacity.

As further cooling may occur in the distribution mains themselves, these should be laid out with a pitch in the direction of air flow so that both gravity and air flow will carry water to drain legs located at appropriate intervals. These should be fitted with automatic drain valves to prevent them becoming flooded.

Any downloops in the distribution mains should be avoided if possible but if this cannot be arranged then drain legs should be located at the downloop.

Except for the drain legs, all take off points from the distribution mains should be taken from the top of the main to prevent liquid water from entering the take off lines — see Fig 3.

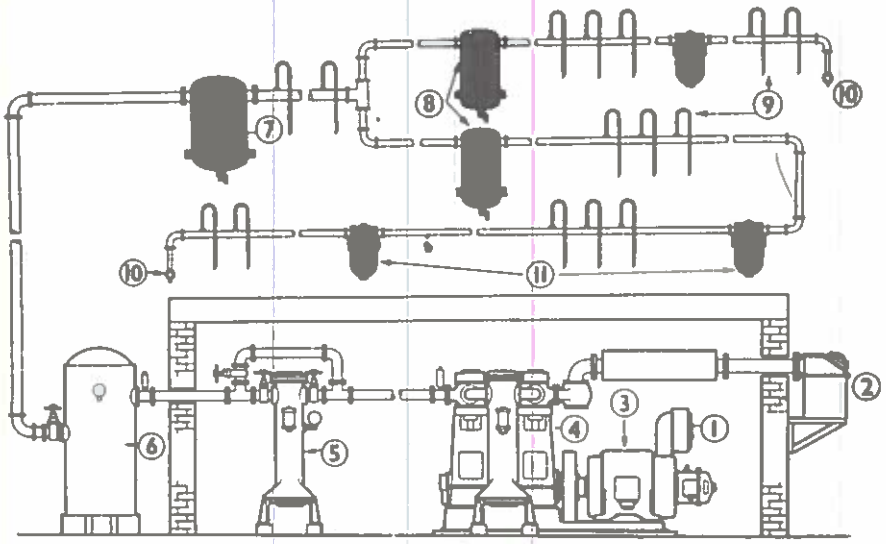


Fig 3
1. Dry fabric filter protecting electric motor cooling air supply. 2. Air filter protecting compressor intake. 3. Electric motor. 4. Compressor. 5. Aftercooler. 6. Receiver. 7. Main pipeline filter/separator. 8. Smaller main filters in branch lines. 9. Supply points. 10. Drain cock. 11. Individual air line filters. (Vokes Limited).

Automatic Drains

Various types of automatic drain valves have been produced which automatically release water collected from the air line when filled. A ball float type valve is commonly used since this maintains a positive shut-off, opening only when enough water has collected to lift the float, and closing again immediately the water has been ejected. The other main type is of 'aerodynamic' design which blasts open to eject the contents of the trap when a specific quantity of water has collected. The main difference between the working of the two types is that the float valve type produces a continuous discharge, and the blast type a heavier discharge at intervals.

domnick hunter filters

STEAM FILTERS
BIO - X ELEMENTS
OIL - X COMPRESSED AIR FILTERS - FLANGED RANGE
OIL - X STAINLESS STEEL RANGE
MEDICAL VACUUM FILTERS
HIGH PRESSURE FILTERS
OIL - X ELEMENTS
BIO - X AIR STERILISATION FILTERS - PORTED RANGE
OIL - X COMPRESSED AIR FILTERS - PORTED RANGE
AUTOCLAVE FILTERS
BIO - X AIR STERILISATION FILTERS - FLANGED RANGE
AIR/OIL SEPARATORS
AUTOMATIC DRAINS

quality first and foremost.

domnick hunter filters limited

Durham Road, Birtley, Co. Durham, England DH3 2SF
Telephone: 0632 405121 Telex: 537282

Can you measure the cost of less-than-perfect filtration?

An inadequate filtration system declares its costs all too clearly in bills for maintenance, repairs and replacements.

That is why it is important to be ultra careful whose filter system you choose.

ultrafilter is a specialist filtration company with a unique product, covered by patents in Europe and America and offering an exceptional guarantee.

ultrafilter guarantees its filter housing for 5 years, its OL filter elements for 12 months and its service to be with you in 24 hours.

ultrafilter is committed to the highest standards of filtration achievable and sustains an on-going R & D programme to meet the even more exacting demands of tomorrow.

ultrafilter gmbh Dusseldorf,®
Bussingstrasse 4, D-5657 Haan 1,
Dusseldorf, West Germany.
Telephone: (02129) 2012 Telex: 859508 ufd

ultrafilter ltd.,
33 Sandy Way, Amington Industrial Estate,
Tamworth, Staffs. B77 4DS
Telephone: (0827) 58234 Telex: 342323

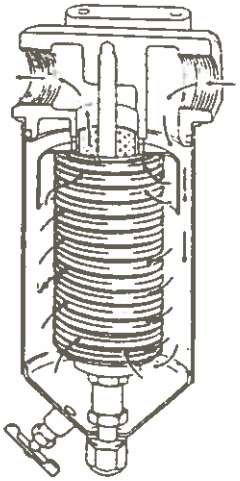
BRITISH PATENT NOS.
1555847 1557946 1595684

ultrafilter UNIQUE Guarantee
* 5 YEARS for filter housing material
* 12 MONTHS on OL elements
* 24 HOUR service

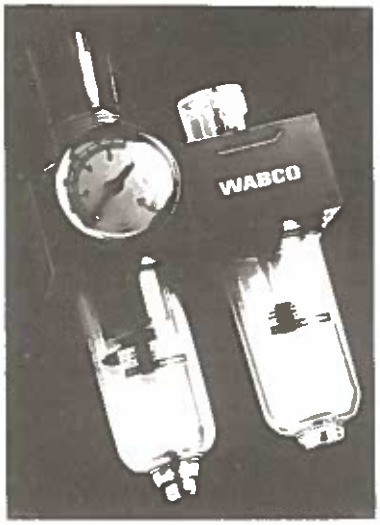
Under pressure. Perfect filtration.

ultrafilter

Two limitations may apply to mechanical drains (or traps) — the possibility of becoming 'air locked', and limited ability to handle water contaminated with oil. The former condition may be countered by fitting a pressure balance pipe. Where oil contamination results in a build up of emulsion which makes the automatic drain sluggish in operation (or even inhibits satisfactory operation), more frequent cleaning is necessary, or a modified form of valve, capable of handling heavy contamination should be used. Some types of mechanical traps may also be susceptible to blocking or jamming open in the presence of scale or larger solid contaminants, although many are designed to have self-cleaning valve actions.



Felt element air line filters (Air Maze Ltd).



Combined line filter, separator and pressure gauge. (Wabco Westinghouse)

Line Filters

Because of the possibility of cooling occurring during the passage of air through the distribution mains, it is preferable to install individual smaller filters as near as possible to the actual point of use of the air rather than a single large filter adjacent to the air receiver. These filters should also be located upstream of any pressure reducing valves.

The fact that the aspired air is filtered, and that a proportion of any solids carried through the compressor will be withdrawn from the airstream by the separator, does not eliminate the necessity for filtering the delivered air via line filters. These can, in any case, provide a higher degree of protection than an intake filter without excessive pressure drop, as well as taking care of any contaminants originating in the compressor. Thus, a large system may require additional high flow rate filters in the main and branch main lines, as well as individual line filters. Alternatively, filters may be used in individual lines for specific protection of individual appliances or devices. The latter has the advantage that pressure is dropped only through a single filter to each take-off point rather than subjecting the main line flow to pressure drop through filters. On the other hand, main line and branch line filters can reduce the number of individual filters required in the system.

A properly designed air line filter of the correct size for the rate of air flow will effectively remove the liquid water, but cannot reduce the water vapour content of the air. If the air is

subjected to further cooling after it has passed through the filter more water may condense out. If complete freedom from water contamination is essential then steps must be taken to ensure that the water vapour content of the air is lowered to the point at which the 'Dew Point' of the air is lower than any temperature than can be attained by the air itself.

Standard air line filters for distribution lines generally remove particles down to about 40–50 μm in size, or lower in some cases. Finer filters can be used for better protection where required, but it is more usual to use these for second stage filtering at the take-off point to individual supplies. In this respect line filters fall into four categories, viz:

(i) 'Roughing' filters for distribution mains, capable of removing particles down to 50 μm.

(ii) 'Medium' efficiency filters capable of removing particles in the range 5–40 μm.

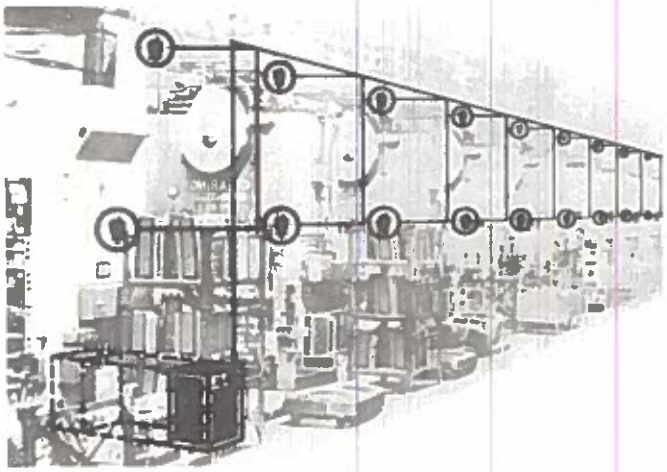
(iii) 'Fine' filters capable of removing particles in the size range 1–5 μm.

(iv) 'Ultra-fine' filters capable of removing particles down to 0.1 μm or better.

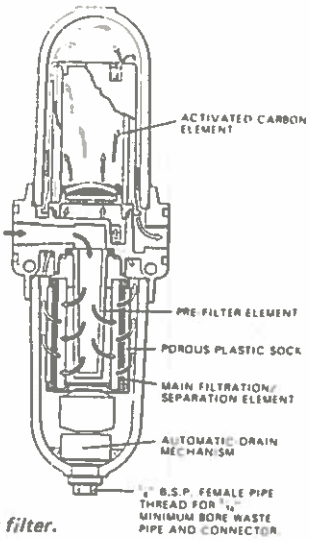
Fine and ultra-fine filters are normally only used for second and third stage filtering, respectively, *ie* should be preceded by a coarser filter to remove coarser contaminants and protect the finer filter from gross dirt contamination. Some specific recommendations for the filter rating required are given in Table III.

Oil Removal

When the air is compressed in a lubricated compressor small amounts of oil will inevitably leave with the compressed air. Since this oil has been subjected to high temperatures during air compression it will have been oxidized and will no longer be suitable as a lubricant – hence the necessity of injecting a separate oil supply into a system that needs lubricant. However, the amount of degraded oil leaving a modern lubricated compressor is relatively low and partially separated out in the aftercooler anyway. The small residual of degraded oil is normally quite acceptable for general industrial systems; or equally can be removed in sufficient quantities by normal air line filters.



Parker 'Auto-fill' oil mist lubricator fed from a centralized oil reservoir.


ACTIVATED CARBON ELEMENT
PRE-FILTER ELEMENT
POROUS PLASTIC SOCK
MAIN FILTRATION/ SEPARATION ELEMENT
AUTOMATIC DRAIN MECHANISM
B.S.P. FEMALE PIPE THREAD FOR MINIMUM BORE WASTE PIPE AND CONNECTOR

Norgren oil-removing filter.

TABLE III – SUGGESTED DEGREE OF CLEANLINESS REQUIRED
((IMI Norgren Limited)

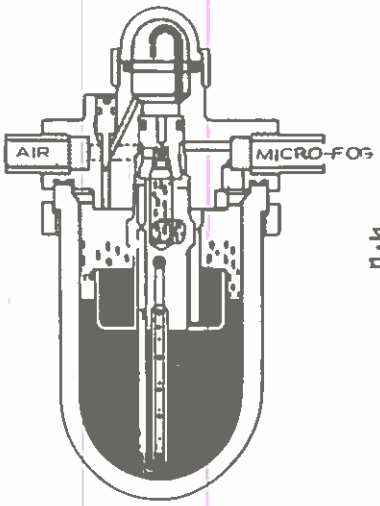
Type of Industrial Process	Water Removal		Oil Removal		Particulate Matter Removal			
					Above	μm		Below
	Liquid	Vapour	Aerosols	Vapour	40	10–25	3–6	1
Air agitation	E	C	E	E	NS	NS	E	C
Air bearings	E	C	E	X	NS	NS	E	C
Air conveying and flotation								
Foodstuffs	E	E	E	E	NS	NS	E	C
Powders	E	E	E	C	NS	NS	E	C
Granular products	E	C	E	X	NS	E	C	X
Air motors: rotary, reciprocating, Linear (air cylinders)								
Heavy duty	E	X	X	X	E	X	X	X
Light duty	E	X	X	X	E	X	X	X
Miniature	E	X	X	X	E	C	X	X
Ultra high speed (above 100 000 rev/min)	E	X	X	X	NS	NS	E	X
Blow guns, cleaning and cooling nozzles								
Cleaning of electronic equipment	E	E	E	C	NS	NS	E	C
Cleaning of food or pharmaceutical containers	E	E	E	E	NS	NS	NS	E
Cleaning of machine parts	E	X	X	X	NS	E	X	X
Cooling of glass and plastics	E	C	E	X	NS	NS	E	X
Cooling of metals	E	X	X	X	E	X	X	X
Film processing	E	E	E	C	NS	NS	NS	E
Breathing masks and protective clothing	E	C	E	E	NS	NS	NS	E
Fluidics								
Input sensors, logic circuit, output devices	E	X	E	C	NS	NS	E	X
Power circuits	E	X	X	X	E	C	X	X
Food, drink and tobacco processing	E	E	E	E	NS	NS	NS	E
General machinery								
Boot and shoe machines	E	X	X	X	E	X	X	X
Brick, clay and tile machines	E	X	X	X	E	X	X	X
Foundry machines	E	X	X	X	E	X	X	X
Glass machines*	E	X	X	X	E	X	X	X
Laundry and dry cleaning machines*	E	C	C	X	E	X	X	X
Machine tools	E	X	X	X	E	C	X	X
Packaging, paper and printing machines*	E	X	C	X	E	X	X	X
Shot blasting plant	E	C	X	X	E	X	X	X
Textile machines*	E	X	C	X	E	X	X	X
Welding machines	E	X	X	X	E	C	X	X
Instrumentation								
Pneumatic gauging equipment	E	X	E	X	NS	E	C	X
Precision pressure regulators	E	X	E	X	NS	NS	E	X
Process control instruments	E	C	E	X	NS	E	C	X
Pneumatic circuits and valves								
Directional control, flow control pressure control valves excluding precision type	E	X	X	X	E	X	X	X
Spray guns†	E	C	E	X	NS	NS	E	C
*If air comes into contact with product	E	C	E	X	NS	NS	E	X

Key:

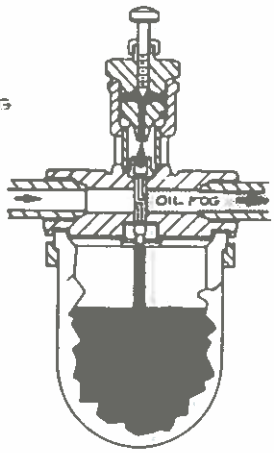
C – should be considered; E – essential; NS – not suitable; X – not normally required

† Size of particulate matter should not exceed 50% of the thickness of the paint layer in order not to roughen the finish.

System of oil feed to Parker 'Auto-fill' lubricators



Micro-fog lubricator (Norgren)



Oil fog lubricator (Norgren)

Certain systems demand pure, oil-free air, in which case the choice lies between the use of an oil-free compressor or further treatment of the delivered air. The former is not necessarily a complete solution. Oil-free compressors still produce compressed air contaminated with dirt and water and it can be more economical to use lubricated compressors, in conjunction with aftercoolers and standard air line filters, and to fit special Ultra High Efficiency filters to remove oil only from those parts of the compressed air system where oil free air is essential.

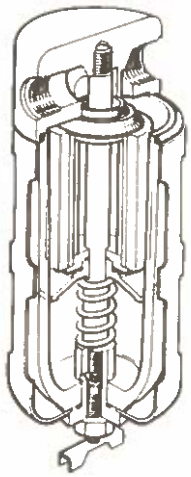
The problem of oil removal is complicated by the fact that oil in compressed air can exist in three forms — liquid oil, oil/water emulsions and oil vapour. Special filters are required to remove oil vapour and oil aerosol.

Coalescing Filters

Modern oil removal filters are of the coalescing type commonly using glass fibre elements. Oil particles of varying sizes impinge on and adhere to the fibres, resulting in a gradual build up of coalesced droplets. These droplets are driven to the outside of the filter by the air stream. When the oil comes to the outside of the material it is stopped in a porous sock covering the element. It then flows by the force of gravity down to the bottom of the sock where it drops to the filter bowl. The oil is then automatically drained from the filter. These filters are capable of removing the oil content in the compressed air down to a level of 0.1 mg/m^3 or less.

In addition to removing oil droplets, such filters have an equally high efficiency in removing minute water droplets but they should be protected against gross dirt or water contamination by means of standard air line filters mounted upstream.

The critical oil particle size is 0.1 to 0.5 μm droplets. Some manufacturers state 0.3 μm but the figure is slightly dependent on the kind, viscosity, temperature and quantity of the oil particles fed to the filter. Filters capable of retaining this size will also retain any other sized droplets. Smaller particles will be captured by the fibres due to their Brownian movements and the bigger droplets mainly by inertia effect.



Spirax — Monnier coalescing filter

TABLE IV — TYPICAL SIZING OF COALESCING FILTERS FOR OIL REMOVAL

Nominal Port Size		Air Flow at 6.3 bar gauge
ISO/7	ISO/R228	dm^3/s A.N.R.
R_c 1/8	G 1/8	up to 1
R_c 1/4	G 1/4	up to 4
R_c 3/8	G 3/8	0.25 to 8
R_c 1/2	G 1/2	1 to 22
R_c 3/4	G 3/4	3 to 35
R_c 1	G 1	5 to 55
R_c 1.1/4	G 1.1/4	7.5 to 80
R_c 1.1/2	G 1.1/2	10 to 100

It is advisable to ensure that these prefilters, fitted in the line ahead of the Ultra High Efficiency filter, are capable of removing dirt particles down to 5 μm or less as otherwise the coalescing filter may quickly become choked with dirt.

As a matter of principle high efficiency oil filters should also always be installed downstream of a dryer, where used. A high concentration of water in liquid form will not only give a higher load to the filter; an element saturated with water will even tend to allow easier passage of oil particles.

As with standard air line filters, the size of the Ultra High Efficiency filter should be selected in accordance with the air flow through it — see Table IV.

Under normal operating conditions the continuous rate of air flow through the filter should not be permitted to exceed 75% of the maximum rated capacity of the filter, as excessive continuous flow results in excessive air velocity with a risk of reintroducing oil into the air stream.

Coalescing filters must be Ultra High Efficiency filters to accomplish their purpose and although standard filter tests exist there is no accepted standard laid down to establish the efficiency of a coalescing filter in removing oil.

Removal of Oil Vapour

For most processes the removal of oil vapour is unnecessary since, unlike water vapour, oil vapour exists only in minute quantities and is not objectionable except in those circumstances where its odour is unacceptable.

For certain processes in the food, pharmaceutical and beverage industries, or where air is being used to supply breathing masks, it is strictly necessary to remove oil vapours. This is most commonly done by passing the air through an adsorbing bed, usually of activated carbon, although other materials can be used.

Combination filters incorporating both coalescing elements and adsorbent beds are available to supply Ultra Clean Air which will satisfy the requirements of BS4275 for breathing quality air as far as freedom from oil is concerned, but special filter units are required to remove carbon monoxide or carbon dioxide and odours.

Dry Compressed Air

Removal of water from the compressed air to a level where no new condensation can be formed in the compressed air system (*ie* the dew point is made lower than ambient temperatures) can be an advantage in any industry, and in some can be an essential feature. Examples of the advantages of 'dry' compressed air:-

(i) Supply to air tools making it possible to lubricate them more efficiently.

(ii) The lubrication of all pneumatic components is improved when dry compressed air is used, and servicing intervals are increased.

(iii) The use of dry compressed air in spray painting equipment eliminates any risk of damage to the paint finish from water droplets.

(iv) In blast cleaning units the reliability of the equipment is improved and risks of icing under outdoor conditions are eliminated when using dry air.

(v) In a dry compressed air system there is no corrosion, which can lead to loss of pressure and leaks; there is also no need for draining off the condensed water.

Air drying is also desirable in conditions of high ambient temperatures and local humidity, such as laundries, *etc*, where it may otherwise be difficult to secure sufficiently low compressed air temperatures to provide ideal conditions for air line filters.

Different degrees of dryness (dew points) are required, depending on the use of the compressed air, the ambient temperatures and humidity. Different types of air dryers with varying dew point performances are therefore available.

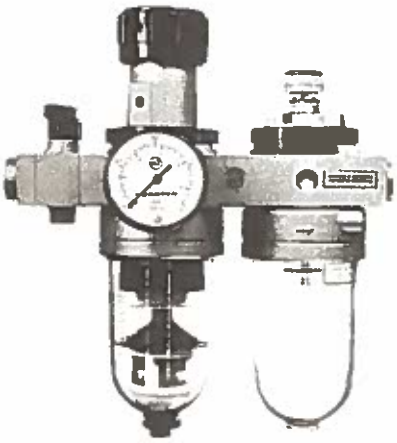
Refrigeration dryers use the simplest drying method; the air passes through an aftercooler but the coolant is maintained at a much lower and constant temperature. The coolant is produced by a closed refrigeration circuit (incorporating compressor, condenser, *etc*) adjusted to cool down the air to a temperature just above 0°C. The limitation for these dryers is the freezing point of water.

Cooling is done in two steps, first in a precooler (cooling with the cold air leaving the dryer) and then in the main cooler (cooling with coolant). All water condensed in these coolers is separated and removed from the air. The lowest obtainable dew point with a refrigeration dryer is approximately +2°C. This dryness is quite adequate for most common uses of compressed air. In fact, by drying air to a dew point which is approximately 10 degC below the lowest possible prevailing ambient temperature in the air system, all risks of getting condensate have usually been eliminated.

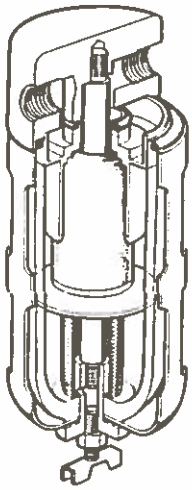
Lower dew points are required when the compressed air can be cooled to temperatures below 0°C, *eg* in compressed air pipes outdoors in winter time. Certain processes, pneumatic instruments *etc* often require dew points below 0°C. For such applications adsorption dryers should be used. With these dryers, dew points down to − 40°C can be obtained.

The adsorption dryer consists of two drying columns filled with desiccant; the compressed air passes through one column where the desiccant adsorbs the water vapour in the air, and this process continues until the desiccant is saturated with water vapour. The air flow is then automatically switched to the second column (with active desiccant) whilst the first column is regenerated and the water removed from the desiccant. Different methods are used to regenerate the desiccant (choice normally being determined by a balance between investment and running costs).

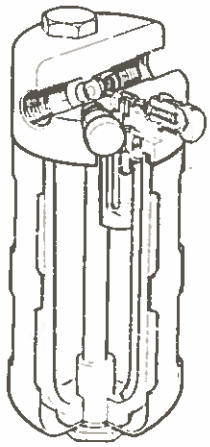
In the heat regenerated adsorption dryer the water vapour is removed by heating the desiccant. The released water vapour is then taken away from the drying column in a current of air. In one type of heat regenerated dryer the heat is generated by external band shaped heaters clamped around the columns. The released water vapour is removed by a small amount of expanded dried



Norgren filter, regulator, lubricator.



Spirax — Monnier compressed air filter



Spirax — Monnier lubricator

air (approximately 1.5%). Large heat regenerated adsorption dryers are regenerated with ambient air sucked in by a blower. This air has been heated up by an external heater before entering the desiccant bed.

The heat regenerated type of adsorption dryer is most suitable for medium to large compressed air installations. For small air flows the so called heatless dryer, which is regenerated with cold air, is more suitable.

The technique of expanding a part of the dried compressed air gives a very dry regenerating air which can remove the adsorbed water from the desiccant. This type of adsorption dryer is of a very simple construction since no heater is required. The amount of compressed air used for regeneration (approximately 15%) limits their use to small rates of consumption.

The sorption dryer is based on a new drying method for compressed air and is so far only employed in a range of dryers specially designed for use together with oil-free screw compressors. The drying is achieved by passing the wet air through a drying material impregnated with a hygroscopic salt.

The drying material is folded into a honeycomb pattern and shaped into the form of a rotor.

Drying takes place by two simultaneous processes:

(i) Adsorption of moisture to the surface of the drying material.

(ii) Absorption of moisture by the hygroscopic salt.

The combination of these two processes is termed sorption. The rotor slowly turns, passing through two separate sections. In three quarters of the rotor section the air is dried and in the remaining section it is regenerated by hot compressed air. As this air has a very low relative humidity, no additional heating is required; therefore, the only power requirement is the small amount of electricity needed to drive the rotor and the control circuit (approximately 0.06 kW). All air used for regeneration is recycled back to the main stream of compressed air.

The dew point after passing through this type of dryer is approximately 50 degC below the temperature of the compressed air entering the dryer.

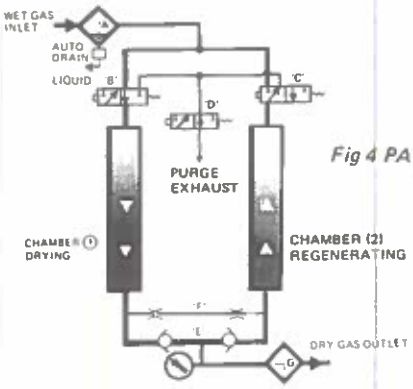
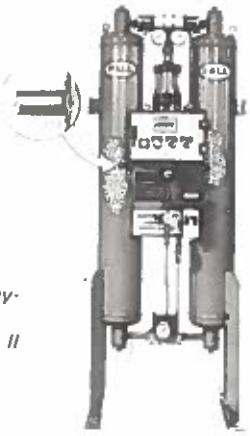


Fig 4 PALL® small flow heatless dryer.



PALL® Heat-Less® Dryer with energy-saving Amloc® II Control. Cutaway shows probe for Amloc II energy saver.

Heatless Dryers

An example of a small flow heatless dryer is shown in Fig 4. Wet incoming gas passes through the reverse Ultipor coalescing filter (A) which removes all liquid contaminants so that the desiccant dryer can operate at maximum efficiency and can perform the function of removing vapour only. Water vapour cannot be removed with a coalescing filter.

The gas, saturated with water vapour, then passes through the switching valve (B) and is directed downwards through chamber (1) where it passes through the activated alumina desiccant. Gas drying takes place by adsorption which occurs when the moisture level in the gas stream is greater than the moisture level of the desiccant. These moisture levels come into equilibrum and, as a result, water vapour in the gas stream condenses as a film on the desiccant. Adsorption is an exothermic (heat releasing) process and, as drying takes place, heat is released and stored in the desiccant bed for use when regeneration is taking place.

Dry gas from chamber (1) then passes through the check valve assembly (E) and the particulate filter (G), which ensures that no particle larger than 0.9 μm passes downstream to the application. tion.

While gas is being dried in chamber (1) the desiccant in chamber (2), which has been wetted in the previous cycle, is being simultaneously regenerated. At the start of its regeneration, chamber (2) is depressurized from the operating pressure in an upward flow direction through valve (C) and (D), and the gas goes out through the purge exhaust. A portion of the dry gas from chamber (1) passes through an orifice assembly (F) and upwards through the chamber being regenerated. This dry purge, with the aid of the heat of adsorption generated during the previous drying period, removes the moisture from the desiccant. The purge carries this moisture through the valve (C) and (D) and out through the purge exhaust. The generation purge flow is countercurrent to the direction of drying for efficient desiccant regeneration.

When the regeneration cycle is complete, the chamber being regenerated is repressurized to full operating pressure by closing the purge exhaust valve (D). Purge gas continues to flow into the chamber until the pressure rises to system pressure. Repressurization before switchover ensures that the desiccant will not be 'jolted' by the inrushing downward gas flow and thus prevent desiccant attention (dusting).

When chamber (2) is fully repressurized the inlet gas is then switched over to chamber (2) through valve (C) for drying. This type of dryer is also capable of removing carbon dioxide from compressed air to a very low level.

Announcing a new air born addition to the Olympian® Plug-In System

The Olympian 10 Series represents an important new addition to the Norgren family. Created to complement the existing universally popular Olympian ranges, the 10 Series provides an extremely neat and very compact plug-in modular system for 1/8, 1/4 and 3/8 installations.

These new high performance, individual filters, pressure regulators, filter regulators, air line lubricators and selected combinations meet your exact requirements with unbeatable simplicity.

Plug-In design virtually eliminates maintenance down-time and the famous Olympian concept minimises spares requirements for multi-tube use throughout your plant.

1/8 pressure gauge and auxiliary ports are incorporated in the yoke design.

Backed by a highly experienced world-wide sales and service organisation, the 10 Series is a proud addition to the Norgren range, giving internationally the finest possible selection of compressed air processing equipment.

For full details write to:

®IMI Norgren Enots Ltd.
Norgren Works, Shipston-on-Stour, Warwickshire
CV36 4PX. England. Telephone: 0608 61676. Telex: 83208.

IMI

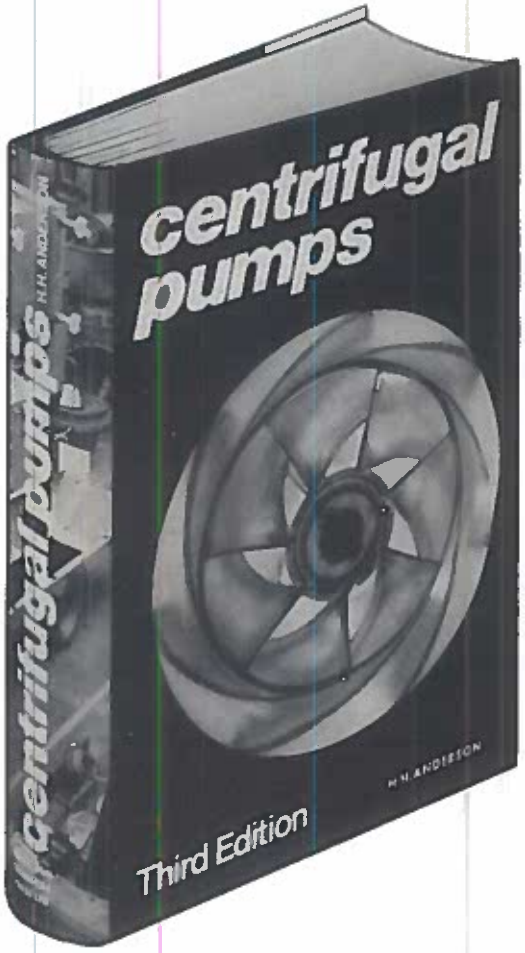
JUST PUBLISHED!

CENTRIFUGAL PUMPS — 3rd EDITION

by H. H. Anderson, BSc(Hons), CEng, FIMechE, Fellow, ASME

This new work by Anderson, the well-known authority on Centrifugal Pumps, whilst written for the user, deals fully with the subject from general principles to a preview of possible developments in the next thirty years.

Contents include: Units of the Système International (SI); General Review of Pumps; General Hydrodynamic Principles; Pump Performance and Dimensions; Theory of Centrifugal Pumps; Type number, Flow and Efficiency; Statistical Analysis of Pump Performance; Entry Conditions and Cavitation; The Determination of Pump Speed; Precision Manufacture of Hydraulic Passage Surfaces for Fluid Machines; Pump Type Variations; Planning a Range of Pumps; Losses in Centrifugal Pumps; Normal Construction Materials; Special Construction Materials; Mechanical Design of Centrifugal Pumps; Critical Speeds and Vibrations; Bearings; Pressure Vessel Aspects; Shaft Sealing; Approach and Discharge Conditions; Characteristics of Pumps and Pipe Systems; Installation of Pumps; Starting, transient and emergency conditions; Testing and Test Codes; Maintenance of Pumps; Single Stage Pumps; Multistage Split Casing Pumps; Borehole Pumps; Submersible Pumps; Cone Flow and Axial Flow Pumps; Multistage Cellular Pumps; Severe Duty Pumps; Small High Speed Pumps; Boiler Feed Pumps; Large Single Entry Pumps; Pumps for Storage of Energy and other Large Power Duties; Mine Pumps; Thermal Power Station Pumps; Offshore Marine and Aerospace Pumps; Chemical Refinery and Process Pumps; Pumps for Various Specific Duties; Drivers; Miscellaneous Aspects; The Third Millennium; Appendix A — Geometry and Performance; Appendix B — Materials; Appendix C — Design Example; Appendix D — Quality Assurance. Appendix E — Water Turbines.



- OVER 500 PAGES,
- 250 DIAGRAMS, ILLUSTRATIONS, TABLES, ETC.
- HARD CASE BOUND
- GOLD BLOCKED, WITH DUST COVER

ORDER YOUR COPY **NOW**

TRADE & TECHNICAL PRESS LTD., CROWN HOUSE, MORDEN, SURREY SM4 5EW, ENGLAND.

A further system is shown in Fig 5, specifically designed to control the drying period on larger heat-activated dryers so that the full drying capacity of the desiccant is used. Extending the designed drying period saves energy and consequent operating costs since there are fewer regeneration periods.

During regeneration this system also ensures that only the correct amount of energy necessary to desorb all the water vapour that has been adsorbed during the drying period is used. The amount of energy required for regeneration is directly proportional to the quantity of water vapour adsorbed by the desiccant during a drying period; there is, therefore, no wastage.

This system controls two functions in gas drying operation: (i) drying time; (2) regeneration. The system consists of a probe placed in the desiccant bed near the outlet to sample the air passing through the drying chambers (see Fig 5A left chamber), a moisture-sensitive cell, and a solid-state circuit.

Air passes through the probe in the desiccant bed to a moisture-sensing cell that has a high electrical resistance value when dry and a lower value in the presence of moisture. When wet air from the drying chamber passes over the cell, the Amloc circuit senses this lower resistance value and switches the air-stream to the other chamber to continue drying.

The control switches drying chambers only when moisture in the air to the probe is detected, thereby using the full adsorption capacity of the desiccant.

The second function of the system is to control regeneration. (See Fig 5B, right chamber). During regeneration, air is heated and passed into the wet desiccant bed to carry the moisture out of the chamber to the atmosphere or to a condenser where the moisture is drained away, thus regenerating the desiccant for the next drying period. The chamber is dry (regenerated) when the temperature of the heated purge exhaust air reaches a predetermined value. A temperature switch is placed in the dryer chamber to sense the temperature of the heated purge-exhaust stream leaving the regenerating chamber. This switch senses this temperature and shuts off the heater(s) (Fig 5C, right chamber), terminating the heating portion of the regeneration cycle.

At this point, the cycle is complete: the system shuts off the purge air supply and the chamber remains on stand-by until the system senses wet air in the drying chamber and signals that switch-over is required (see Fig 5D, left chamber). The dryer then switches over. The wet chamber

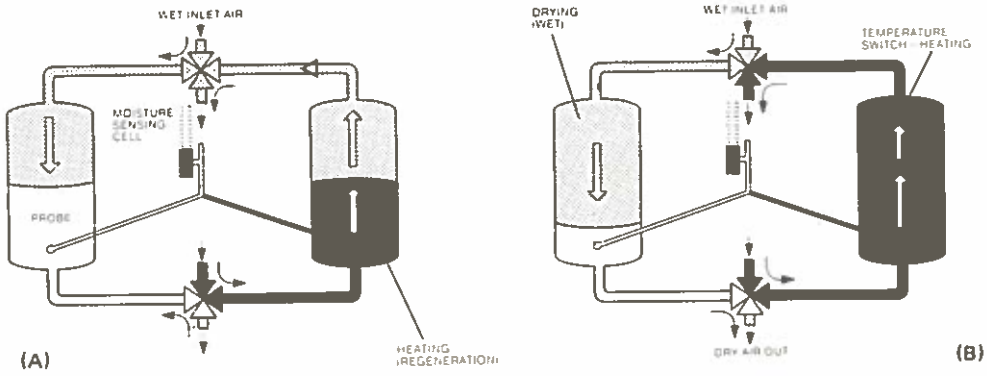


Fig 5 PALL® large flow heatless dryer system (diagrammatic). *(contd) ...*

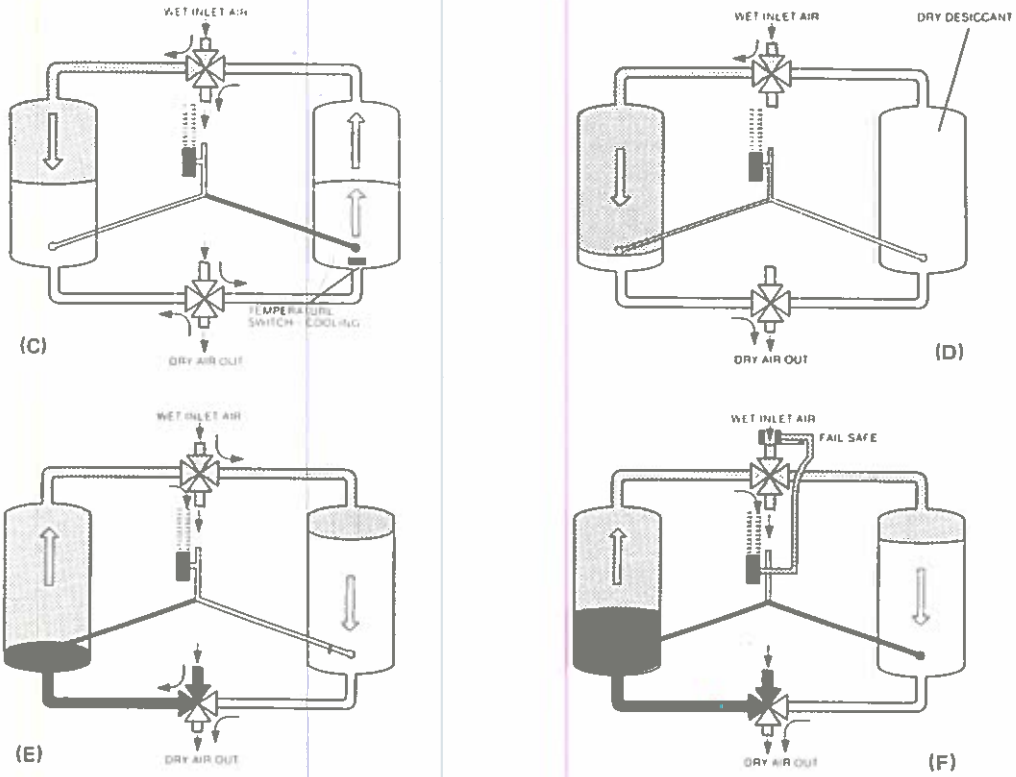


Fig 5 PALL® large flow heatless dryer system (diagrammatic) (contd.)

immediately progresses through regeneration as previously described. (See Fig 5E). The cycle repeats as necessary.

Self-Checking Feature

The system checks its own performance (see Fig 5F) during operation. Periodically, wet inlet air is passed over the moisture-sensitive cell and the solid state circuit measures the electrical resistance change in the cell. With the system operating correctly it continues to control the dryer. If a malfunction should occur in the system, dryer operation will be automatically placed in a fixed cycle and a warning light will alert the user. The moisture-sensing cell can be changed at customer convenience. In any event, dry air continues to pass downstream.

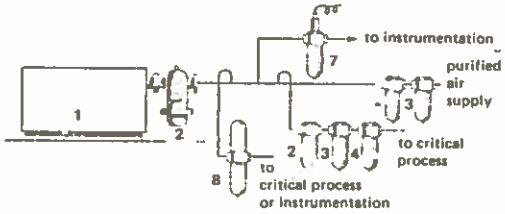
Bacteria Removal

Certain processes using compressed air require the air to be free from bacteria and other micro-organisms, *ie* they require *sterile air*. Such micro-organisms inducted by the compressor are not killed by the heat of compression, so remain in compressed air and can pass through conventional filters. It is, however, possible to trap them in special air sterilization filters using borosilicate microfilm media; or kill them by sterilization under heat. The former method, which is pure filtration, is sometimes known as cold-sterilization.

Installation Recommendations

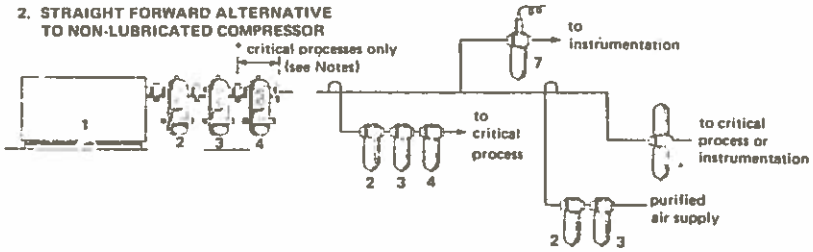
The diagrams and information shown are intended as a general guide. Readers are recommended to obtain specific advice on each installation being contemplated.

1. NORMAL RING MAIN PROTECTION

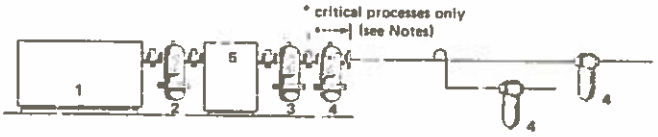


1. Oil lubricated screw compressor with built-in aftercooler.
2. Grade A or PF line filter.
3. Grade AA line filter.
4. Grade AC adsorption filter for extremely critical process applications.
5. Refrigeration dryer.
6. Twin tower desiccant dryer.
7. Combined filter/heater. Combats condensation in instrumentation lines.

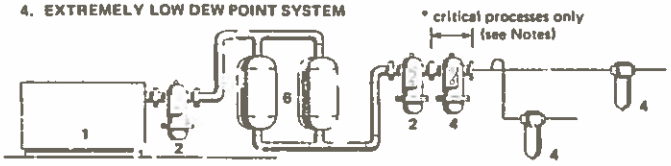
2. STRAIGHT FORWARD ALTERNATIVE TO NON-LUBRICATED COMPRESSOR



3. REDUCED DEW POINT SYSTEM



4. EXTREMELY LOW DEW POINT SYSTEM



Notes

In piston compressor installations it is normal to fit a receiver immediately downstream of the aftercooler.

In rotary vane and piston compressor installations the use of an efficient aftercooler sited immediately downstream of the compressor is essential.

In systems 1 and 2 grades AA and AC filters are added as terminal filter/separators to remove water and oil-vapour which may have condensed due to a fall in temperature of the compressed air as it passes through the ring main. The filter/heater is shown as a less expensive, low maintenance alternative to a refrigeration or adsorption instrument air dryer.

In systems 2, 3 and 4, a grade AC main line filter should be installed only when large volumes of process air are required for a critical process. Wherever possible grade AC should be used as terminal filters.

In system 4 shown above a grade A filter is used to trap desiccant dust introduced into the air stream by the dryer.

It is important that all terminal filters are installed as close as possible to the process or plant using the purified air.

Untreated mild steel pipe work is not recommended for ring mains. Rust and pipe scale can break free from the inside of untreated pipes and reduce the service life of filter elements.

Air Line Fittings

FOR AIR mains larger than 65 mm (2½ in) bore welded fittings are preferred although cast iron pipes with integral flanges or steel pipes with welded-on flanges may be used. Smaller mains in steel pipes are normally screwed to proprietary malleable iron tube fittings, to BS143 and BS1256. ABS pipe is an alternative material for air mains. This material is not suitable for threading. Matching fittings in ABS are available covering a range of sizes from 16 mm (5/8 in) to 110 mm (4.3/8 in) nominal o.d., joints being completed by solvent welding. ABS pipes may be jointed to metallic components using adaptors or composite unions together with jointing compounds where necessary.

Screwed connections are generally restricted to metallic tubes of larger diameter and thicker walled sections. Taper threads (*eg* BSP taper or API taper) are usually preferred. The best seal is given by having the taper on the male rather than the female part of the joint. These can assemble with either taper or parallel internal (female) threads. Parallel external (male) threads are less suitable for tight joints, except in 'longscrew' form. Longscrew threads, used for joints, incorporate a soft material which is compressed onto the surface of the external thread by tightening a backnut against a socket.

Solderless connections are widely favoured for pipe sizes up to 19 mm (¾ in) diameter. The simple olive joint is effective (Fig 1) provided the tube projects well through the olive. Tightening

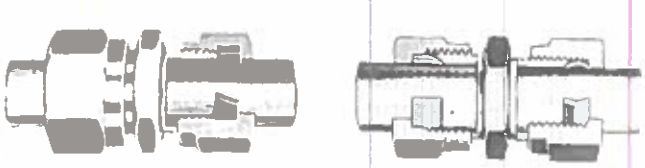


Fig 1 Metal compression couplings with conventional olive (left) and bi-cone olive (right) (Drallim).

Fig 2 Tube coupling with resilient seal in union nut. (Drallim).

the nut then produces a tight compressive lock without deforming the tube wall, this being particularly suited to rigid tubing. Another type of proprietary coupling is shown in Fig 2 which employs a resilient seal.

Various other types of compression fitting are used. Where the tube material is ductile, a collet may be used, tightened up by a nut, to produce an annular deformation of the tube wall to lock the tube within the coupling. The same type of fitting can be used with nylon and plastic tubes, only in this case, support for the inner wall may be provided by an additional inner sleeve – Fig 3.

Fig 3 Stages of assembly of metal compression coupling used with plastic tube. (Drallim).

Nylon tube and PVC hose are now widely used for branch and service lines up to about 12.5 mm (½ in) diameter, and less frequently in larger sizes (up to 25 mm (1 in) in diameter). In addition to permament installations, traditionally piped in metallic tubing, nylon or PVC tubing are particularly suitable for connecting lines where slight movement may be experienced.

Apart from their lower cost and ease of manipulation, nylon and PVC tubing have other advantages over metallic tubing, particularly the absence of fatigue and freedom from corrosion. They are, however, more limited in their service temperature range, and are not suitable for pressure rating at temperatures in excess of 90–100°C. The actual softening point of nylon is of the order of 140–150°C with an internal pressure of 60–70 lb/in^2 (4.2–5 bar), reducing to circa 50°C with an internal pressure of the order of 250 lb/in^2 (17.5 bar).

Compression couplings in brass are the usual choice for nylon and PVC tubes, being non-corrosive. Threads are normally BSP. Hose connectors and screw couplings for heavier duty are normally made in hardened steel. This also enables the wall thickness to be reduced, maintaining a full flow area with minimum pressure drop.

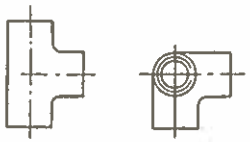
Connectors normally consist of a hose jointing nipple with the hoses secured in place with clamps or clips; or screw connectors with a cap nut and adaptor matching the taper of the hose nipple.

A variety of quick-release couplings are also produced. These are widely used where, for example, one air tool has to be changed for another at a single take-off point.

Various designs of 'instant' couplers have also been produced for use with nylon tubes. The simplest is a barbed fitting over which the tube is pushed, the barbs restraining the tube against outward movement. Other types employ various forms of locking to hold the tube secure once pushed in place. Nylon tube can only be removed from a barbed coupling by cutting. With other 'instant' fittings, however, it may be possible to remove and replace the tube many times over, if required.

SUMMARY OF AIRLINE FITTINGS

TYPE		REMARKS
Compression Couplings		Used with copper pipes, nylon tubes and PVC hose. Usually in brass. Numerous proprietary designs.
Screwed Couplings		Screwed fittings used with metal pipes. May be in copper alloy, cast iron or hardened steel. Male and female BS143 and 1256:1968 applies. Screw couplings for hoses incorporate a nipple
Hose Connectors		Normally comprise an adaptor (connector) hose nipple and cap nut. May be in brass or hardened steel.
Quick Couplings		Designed for quick and simple connection/disconnection of hoses to air outlets.
Claw Couplings		Hose couplings which mate regardless of nipple and thread size. Usually in hardened steel.
Nipples		For simple connection of hose lengths. Used with hose clamps.
Banjo Coupling		Compact type of fitting for connection or take-off at 90 degrees.

Bulkhead Connector		Used to provide line or hose connection through a bulkhead or partition.
Reducers		Provide interconnection between two pipes or hoses of different sizes. May be straight, tee, V- or other configurations
Elbows		Connectors for radiused 90 degree or 45 degree bends used with rigid pipes. May be plain or side outlet. May be equal or reducing. Twin elbows provide connection for three lines.
Tees		T-shaped connectors for connecting three lines. May be equal or reducing, male or female threaded.
Y-Fittings		For direct connection of two separate hoses to a common line — alternative to a Tee.
Cross (4-way) Connector		+ shaped connection for connecting four lines. May be plain or reducing.
Return Bends		180 degree open bends, or with back outlet return
Plugs and Caps		Fittings for sealing the open ends of pipes.

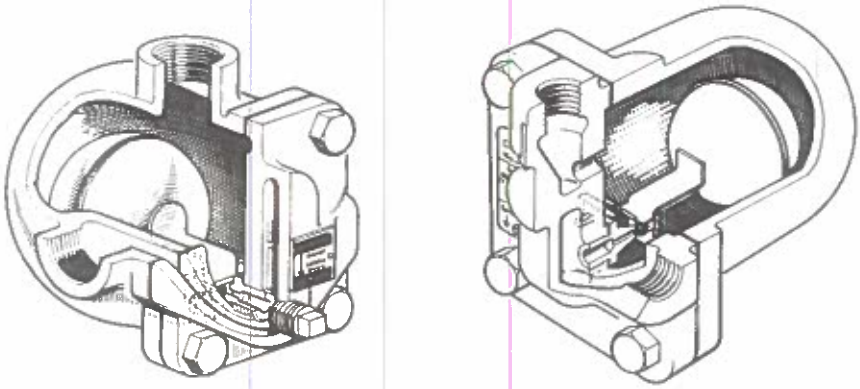


Fig 4 ball float type compressed air drain traps.
(Spirax Sarco).

Main Line Traps

Separators may be fitted in distribution mains for removal of suspended water droplets, preferably associated with an automatic drain. A ball float trap is commonly preferred because of its positive shut-off in the absence of water and positive lift when the trap is filled with water – Fig 4.

Separators may also incorporate filter elements designed to remove solid contaminants and/or oil droplets. Units of this type are more properly called filter/separators.

Various types of automatic drain valves have been produced which automatically release water collected from the air line when filled. A ball float type valve is commonly used since this maintains a positive shut-off, opening only when enough water has collected to lift the float, and closing again immediately the water has been ejected. The other main type is of 'aerodynamic' design which blasts open to eject the contents of the trap when a specific quantity of water has collected. The main difference between the working of the two types is that the float valve type produces a continual discharge, and the blast type a heavier discharge at intervals.

Two limitations may apply to mechanical drains (or traps) – the possibility of becoming 'air locked', and limited ability to handle water contaminated with oil. The former condition may be countered by fitting a pressure balance pipe. Where oil contamination results in a build up of emulsion which makes the automatic drain sluggish in operation (or even inhibits satisfactory operation), more frequent cleaning is necessary, or a modified form of valve should be used, capable of handling heavy contamination.

Some types of mechanical traps may also be susceptible to blocking or jamming open in the presence of scale or larger solid contaminants, although many are designed to have self-cleaning actions.

See also chapter on *Compressed Air Treatment.*

SECTION 2c

Valves

Valves

PNEUMATIC VALVES are generally divided into three main categories defined by function, viz:

(i) Directional control valves.

(ii) Flow control valves.

(iii) Pressure control valves.

In addition there are a number of specialized valve types which do not fall into these general categories, including valves employed as 'logic' elements. (See chapter on *Logic Circuits*).

Directional Control Valves

Directional control valves are categorized by the number of port openings or 'ways', *eg*

2-port (2-way) valve either opens or closes a flow path in a single line;

3-port (3-way) valve either opens or closes alternative flow paths between one or other of two ports and a third port.

Other basic configurations are 4-port (4-way) and 5-port (5-way).

Such valves can also be described by number of *positions* provided; and also whether the outlet is open or closed in the non-operated position, *ie* normally open or normally closed, respectively. Description may also be simplified by giving number of ports and positions as figures separated by a stroke — *eg* a 4/2 valve is a 4-port, two-position valve.

Fig 1, for example, shows a 3-way, two-position directional control valve in simplified diagrammatic form.

The 3-way, two-position valve is thus a logical choice for a single-acting cylinder circuit control or for any other single circuit where downstream air has to be exhausted. It is not necessarily a complete answer to such circuits and there are cases where the use of two 2-way, two-position (straightway) valves may be preferred.

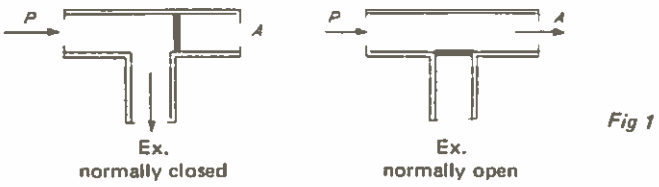


Fig 1

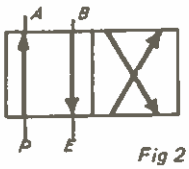


Fig 2

Increasing the number of ports, and if necessary the number of positions, extends the switching capability of a valve. Thus, the 4-way, two-position valve of Fig 2 has two through-connections. In one position P is connected to A, and B can exhaust downstream air through E. In the second position P is connected to B, and A can exhaust downstream air through E. Thus a 4-way, two-position valve can be used to operate a double-acting cylinder or any other device requiring alternate pressure and exhaust on two connecting lines.

Whilst this is a logical control valve choice for a double-acting cylinder, again there are instances where the use of two 3-way, two-position valves may be preferred, *eg* for close coupling or mounting the valves directly on the cylinder ports.

The 5-way, two-position valve of Fig 3 is similar to the four-way valve, except for the provision of an additional exhaust port. Thus, when either A or B is switched to exhaust, it operates through a separate exhaust. This can have specific advantages in particular applications.

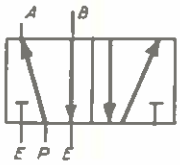


Fig 3

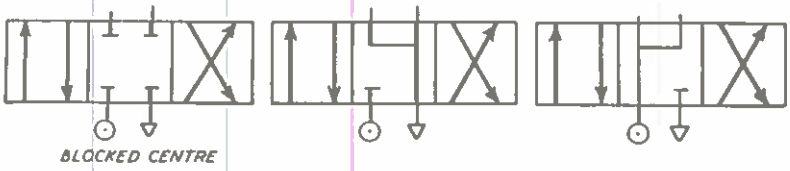


Fig 4

The other directional valve is the 4-way, three-position valve, which can have three possible modes — Fig 4. In the first (left) the function is similar to a 4-way, two-position valve except that an additional position is available with all ports blocked, *ie* no flow is possible through the valve in either direction. This is usually the normal position. A typical application is the control of a double-acting cylinder in which 'hold' facility is also available, *ie* pressurized air can be trapped on both sides of the piston.

In the second mode (centre), conventional 4-way switching is available from the two extreme positions, but the mid position shuts off the supply and connects both downstream lines to exhaust through the valve. Again this type of control is used with double-acting cylinders where it is required to 'free' the piston by exhausting air from both sides of the piston.

In the third mode (right), normal switching is available from the two extreme positions but the centre position provides pressurized air to both downstream lines, with the exhaust closed. This provides an alternative hold facility for double-acting cylinders with both sides of the cylinder continuously pressurized. This will not, of course, give true 'hold' on a single-rod cylinder, unless the pressure is dropped in the line to the blank end side of the piston to compensate for the loss of effective piston area on the rod side.

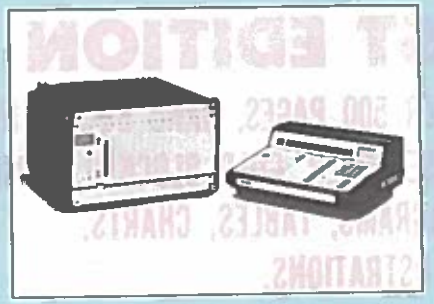
Standard Valve Symbols

Symbols used for designating valves on pneumatic circuits are conventionally in the form of adjacent squares, each square representing one position of the valve. Thus a two-position valve would have two adjoining squares and a three-position valve three adjoining squares. Interior connections between points in each position are then indicated by an arrow or arrows within the appropriate square, together with some indication of unconnected points.

Standard valve symbols adopted through Europe are shown in Fig 5. Additional symbols may also be appended to show the method of actuation — Fig 6. (See also *Data Section* for further symbols).

Industrial control technology demands more and more skilled employees and a continuance of their further training FESTO DIDACTIC is one of the most experienced and efficient participants in creating training systems and programmes for all technical training. We develop, produce and market our systems for pneumatics, hydraulics, mechanics, electrics and electronics, world wide in all the major languages and standards.

FESTO DIDACTIC

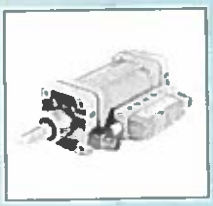


For a quarter of a century, FESTO has been successfully solving control problems. This practical experience and technical knowledge is applied to the conception of freely programmable electronic control system known as the FESTO FPC. The FPC reduced costs, for time is money with English Language programming, error diagnostic system, easy installation and wiring, adaptability and FESTO reliability—in fact, a true partner in productivity.

FESTO ELECTRONIC

FESTO PNEUMATIC

Partners In Productivity



FESTO PNEUMATIC are world leaders in control automation and offer the most complete range of products available in Industrial Pneumatics—Pneumatic Cylinders both single and double acting—Pneumatic Valves, mechanically, manually, and electrically operated

Non return, quick exhaust, flow control and sensors—Cylinder Combinations—Construction Units, and all backed by a full range of accessories.
That is why we are partners in productivity both within our divisions and with you, our valued client.

FESTO PNEUMATIC

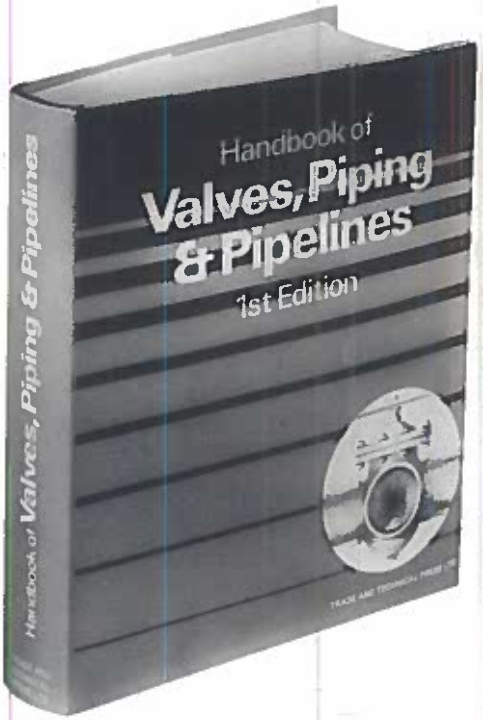
Catherine Wheel Road, Brentford, Middx TW8 8BB
Telephone 01-568 2051/4 Telex 935705

Offices also in Birmingham, West Yorkshire, Scotland and Belfast

HANDBOOK OF VALVES, PIPING & PIPELINES 1ST EDITION

OVER 500 PAGES, HAND BOUND, HARD CASE COVER, GOLD BLOCKED, 2,000 DIAGRAMS, TABLES, CHARTS, ILLUSTRATIONS.

A brand new reference work, now in compilation, to be published early 1982. Contents include:- Section 1 — Introductory:- Piping services, General summary and classification of valves, Valve nomenclature. Section 2 — Types of Valves:- Stop valves, Plug valves, Cocks, Gate valves, Slide valves, Ball valves, Butterfly valves, Globe valves, Rotary valves, Diaphragm valves, Penstocks, Pinch valves, Safety valves, Relief valves. Section 3 — Valves — Design and Construction:- Cocks, Plug valves, Screwdown valves, Gate valves, Gate valves and Globe valves, Butterfly valves, Slide valves, Piston valves, Ball valves, Non-return valves, High pressure valves, Designs for high and low temperatures. Section 4 — Valve Actuators:- Manual, Servo operated, Automatic control, Automated control, Semi-automatic control. Section 5 — Performance:- Fluid flow through pipes, Pipe flow calculations: (i) Fluid transport (ii) Frictional losses (iii) Pressure drop, Steam flow calculations, Valve sizing, Cavitation and erosion, Compressible gas flow in pipes, Leak Detection, Flow of mixtures in pipes, Seismic Analysis, Flow through valves, Piping networks, Noise Control. Section 6 — Pipelines:- Types, sizes and applications of pipelines, Pipe joints and couplings, Troubleshooting and maintenance, Flexible pipes and hoses, Cathodic protection, Corrosion and corrosion treatment. Section 7 — Services (Valves):- General purpose valves, Pressure relief valves, Regulating valves, Sampling valves, Mixing valves, Traps, Foot valves, Level control valves, Blowdown valves, Temperature regulating valves, Throttle valves, Pressure regulating valves, Turbine valves, Automatic control valves, Heated (jacketed) valves, Cryogenic valves. Section 8 — Services (Pipelines):- Oil pipelines, Water pipelines, Irrigation and drainage, Small bore piping, Gas pipelines, Heating services, Domestic services, High temperature services, Low temperature services. Special services, Piping and valve design for Nuclear Power Plants. Section 9 — Materials:- Materials for pipes, Materials for valves, Materials and fluid compatibility, Seals and packings. Section 10 — Guides and Standards:- Valve selection guides, Valve and pipework symbols, National and International valve standards, Pipe standards, Pipe threads, Codes of Practice. Section 11 — Data:- Tables, Nomograms, Charts, Formulae. Section 12 — Buyers' Guide . . . . .



ORDER YOUR COPY

NOW

TRADE & TECHNICAL PRESS LIMITED, CROWN HOUSE, MORDEN, SURREY, SM4 5EW, ENGLAND.

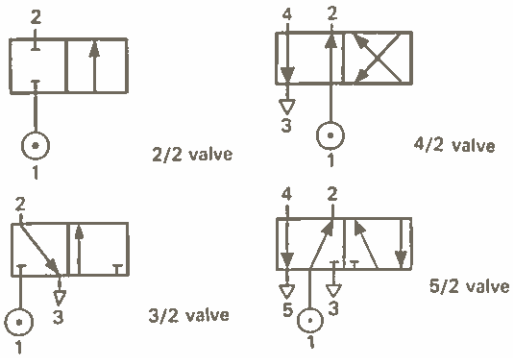


Fig 5

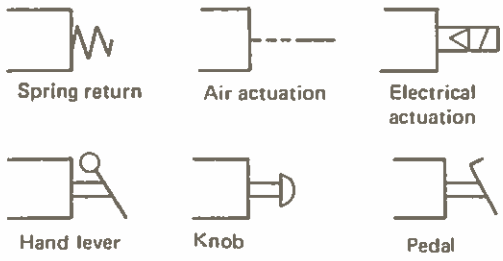


Fig 6

In addition main ports may also be identified by numbers, viz:-

1 = Normal inlet, *ie* main supply port.

2 = Normal outlet port on 3/2 and 3/3 valves.

2 & 4 Normal outlet ports on 5/2 and 5/3 valves.

3 Normal exhaust port on 3/2 and 3/3 valves.

3 & 5 Normal exhaust ports on 5/2 and 5/3 valves.

Port 1 is the main supply port, other odd numbers denote exhaust ports, and even numbers outlet ports. Port 3 is always internally connected to port 2 when the valve is in one position. In the reverse position on a 5 port valve port 5 is always connected to port 4.

Pilot control ports are identified as follows:

'10', '12', and '14'.

Note: Although this numbering is standardized it is not always used. Arbitrary numbering may be used for simplification or convenience in general descriptions.

Valve Sizes

Valve sizes are normally designated by the size of the port tappings. This offers a specific match between pipe size and valve — *eg* a ¼ in valve would have ports tapped to take ¼ in bore pipe tapped externally with BSP or NTP thread. The range of sizes most commonly covered is from 3 mm to 25 mm (1/8 in to 1 in) although main line valves are produced in larger sizes.

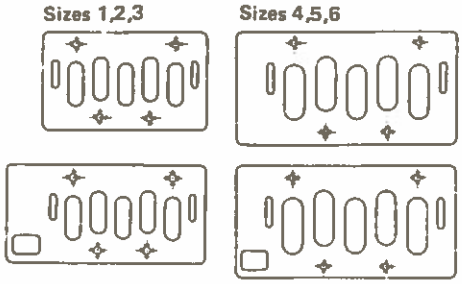
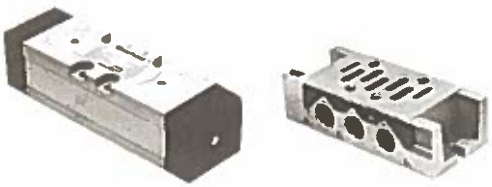


Fig 7 ISO Sub-base patterns.



Sub-base valve and sub-base.

Valve Configurations

The majority of valves used in the UK are of in-line configuration, *ie* designed for direct connection of inlet, outlet and exhaust lines. Continental Europe and America favour the use of sub-base valves, *ie* with valve bodies in block form for mounting on a common baseplate or manifold. All lines are connected to the base.

With sub-base or manifold mounting the valve body normally has no tapped pipe connections. All internal ports are brought to the base of the valve and connection is completed by mounting on a matching sub-base or manifold carrying corresponding ports. Joints in such cases are normally sealed with gaskets or O-rings.

Sub-base mounts are designed to accommodate individual valves. Manifolds are designed to accommodate a number of valves on a common mount.

Connection on the side | For manifold mounting | Connection underneath | Endplate

CETOP

Connection on the side | For manifold mounting | Connection underneath | Endplate

(DIN)

Connection on the side | Endplate | For manifold mounting

(NOMO)

Connection on the side | For manifold mounting | Connection underneath | Endplate for manifold

FESTO (proprietary)

Fig 8 Examples of other sub-base patterns.

TABLE I – FLOW CAPACITIES COVERED BY ISO SUB-BASE SIZES

ISO Number	Pipe Ports Recommended (inch)	C_V – range	k_V – range*	Flow Range (Q_n = l/min)
1	1/8, 1/4	0.5– 1.0	0.4– 0.9	400– 1 000
2	1/4, 3/8	1.0– 2.0	0.9– 1.8	1 000– 2 000
3	3/8, 1/2	2.0– 4.0	1.8– 3.6	2 000– 4 000
4	1/2, 3/4	4.0– 7.0	3.6– 6.3	4 000– 7 000
5	3/4, 1	7.0–12.0	6.3–10.7	7 000–12 000
6	1, 1¼, 1½	12.0–17.0	10.7–15.2	12 000–18 000

*$C_V = 1.12\ k_V$

Port configurations are standardized (*ie* to ISO specification in Europe), thus valves of different manufacture can readily replace ISO valves by another manufacturer on the same base, and vice versa. It does not follow, however, that the actual overall dimensions of the valve or base will be identical under this standard – only the matching port configuration – see Fig 7.

Other standards for sub-base patterns exist – *eg* DIN, CETOP, (NOMO, *etc*) – as well as individual proprietary designs and may be found on existing valves – see Fig 8. They are not interchangeable with ISO or with each other. A particular feature of the ISO pattern is the dimensioning of slots to allow for approximately 50% more flow than the corresponding pipe size so that the flow resistance in the sub-base can be ignored compared to the flow resistance in the valve itself (see also Table I).

Valve Types

Valve types are categorized by principle of operation, the main types being poppet, sliding piston with O-ring seals, shaped spool with static seals, glandless spool, sliding plate and rotating disc.

Poppet Valves

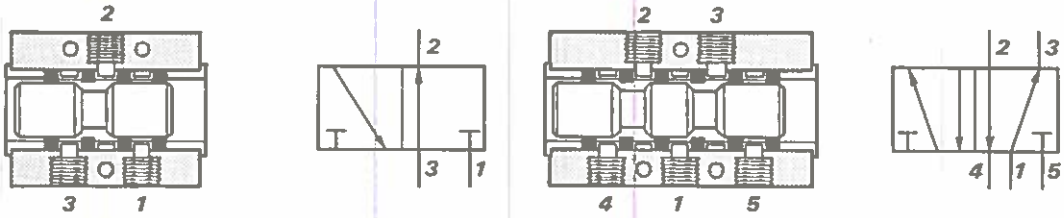
Poppet valves require minimum movements to achieve full opening, offer low resistance to flow when in the open position and have good sealing properties when closed. They can also be made to seal adequately for long periods. They do require relatively large operating forces, however.

A simple poppet valve, designed for a pressure of 10.5 bar (150 lb/in^2) may require an opening force as large as 10 kgf (22 lbf) because of its unbalance. On the other hand, the fact that full opening is achieved with only small travel gives rapid full flow response to signal, which can be of particular advantage in high speed applications and for pilot circuits.

A limitation of poppet valves is restricted switching ability. The basic poppet valve is capable only of on–off switching function. To perform a 3-way or 4-way function, a combination of poppet valves is needed. Other types of valves are thus normally used for such duties.

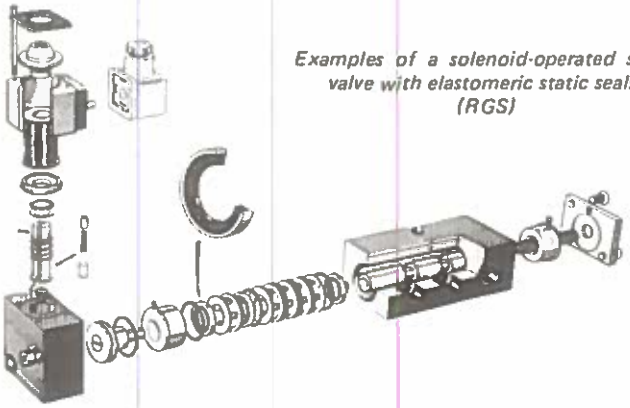
Spool Valves

Spool valves are attractive because they offer extreme flexibility in interconnection via a simple sliding mechanism. They are also relatively simple and economic to manufacture, although for adequate sealing fine surface finish is required on both the spool and barrel bore, with close tolerances to ensure practical minimum clearances. Glandless spool valves thus normally require a lapped fit between spool and body. Spool valves with static seals offer simpler construction in this respect and also rather more flexibility in design with seals positioned between valve spacers so that a seal is situated between each subsequent port and one seal on the outside of each of the two outer ports – see Fig 9.



3-way valve 4-way valve

Fig 9



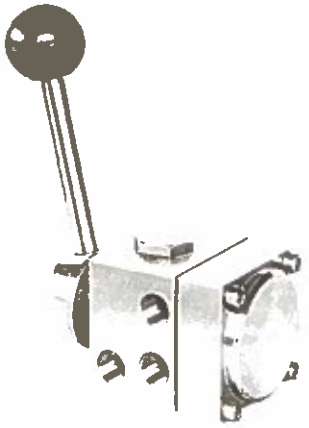
Examples of a solenoid-operated spool valve with elastomeric static seals. (RGS)

Spool valves embody two basic elements, a cylindrical barrel in which slides a plunger. Port blocking is provided by lands or full diameter sections on the spool with intervening waist sections which provide port interconnection through the barrel. This makes it easy to provide multi-way and multi-position switching.

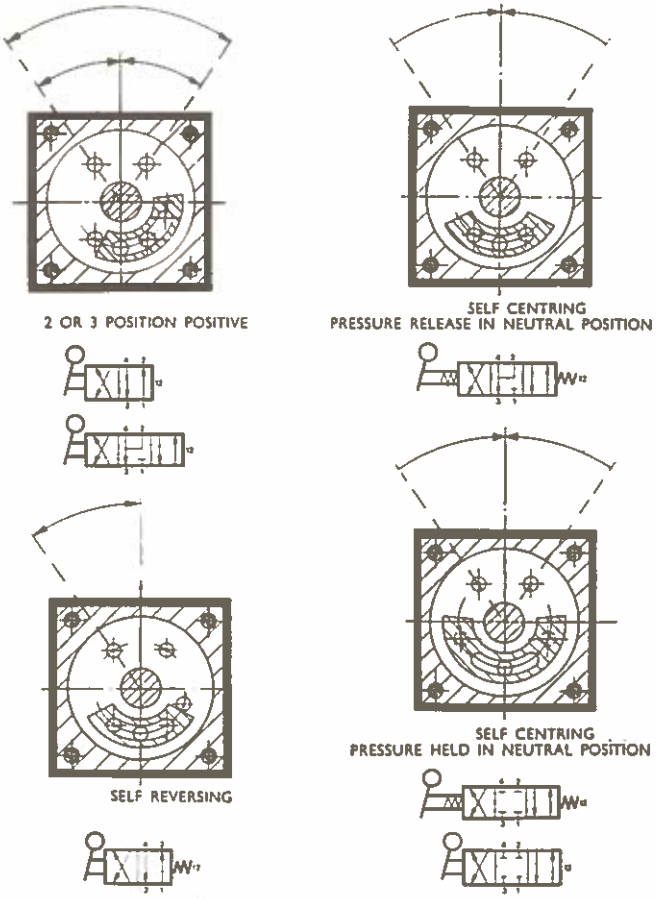
A possible limitation of glandless spool valves is that in certain applications and particularly with larger spool valves, the precision bore and fit may be distorted by externally applied loads. On the other hand, spool valves with static seals also have some limitations. A disadvantage in this case is that the sealing rings must slide over the ports where they may be subject to cutting action or distortion. Variations, primarily again in the case of air valves, include the use of 'U' seals, bushed bodies and ports drilled with a number of fine holes where sealing rings pass, and a linered construction with a liner floated on simple seal rings. All are aimed at reducing the possibility of damage or distortion of the sealing rings sliding over port openings.

Slide Valves

Slide valves offer an alternative approach equivalent in effect to an unwrapped spool. Ports are normally drilled in the fixed slide and matching recesses in the sliding member. The two members are then lapped to fit and held in contact by spring loading. Various methods may be used to reduce friction forces. Thus, in some valves, the sliding member is suspended on spring plates to prevent metallic contact. In other cases, friction forces are reduced by incorporating hydrostatic pressure balance within the valve. The principal limitation of slide valves is that they demand extreme precision of manufacture in order to realize leakage rates comparable with those of other valve types. Slide valves are also sometimes known as plate valves or sliding plate valves.



Semi-rotary valve. (RGS)



Rotary Plate Valves

Rotary valves are another form of plate valve or sliding valve with the moving element rotating over a fixed plate to open or block ports drilled in that plate. Channels cut in the rotary plate can provide transverse flow. Reaction-forced compensation can be introduced by the use of deflector veins to reduce friction. However, all sliding plate valves, *ie* slide valves and rotary valves, are basically restricted to two-, three-, or four-way operation. Rotary valves have the particular advantage that they are readily adapted to multi-outlet working. For this reason they are often favoured as sequence valves in pneumatic circuits. Spool and plate valves are also commonly used in combination in two-stage valves. Two-stage valves are intended to overcome the practical limitations of power response imposed by single-stage operation. A two-stage valve merely employs one valve as a first stage and another at the second stage, normally combined in one body or unit.

An example of a particularly versatile form of rotary valve is shown in Fig 10. Basically this consists of a square or hexagonal section body with four or six peripheral ports communicating with a central chamber. An operating spindle passes through this chamber, operating spring-loaded poppets in individual ports by cam action. A number of valves of this type may be ganged to form a multibank valve.

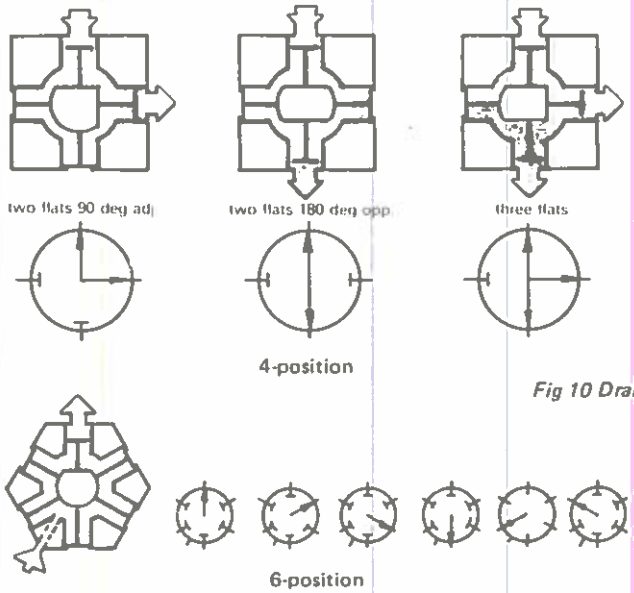


Fig 10 Drallim rotary switch valve.

Shuttle Valves

The shuttle valve is an 'automatic' type of directional control valve with three-way, two-position characteristics. This is based on a simple three-port body with a moving element in the form of the free spool or shuttle, or a ball – Fig 11. A shuttle valve gives an outlet from either one of two pressurized inlets (whichever is the greater value) *ie* the higher pressure inlet moves the shuttle to a position where it blocks the other inlet. It is thus more of a special valve than a true directional control valve and is basically used for logic OR switching.

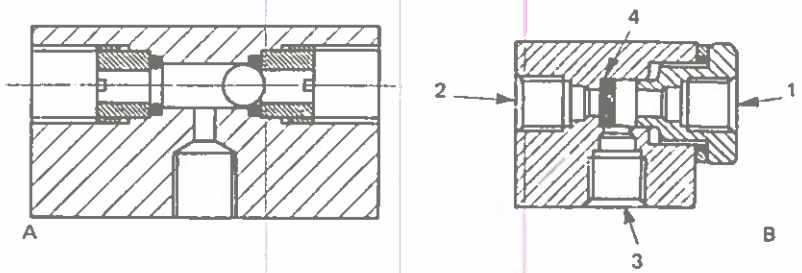
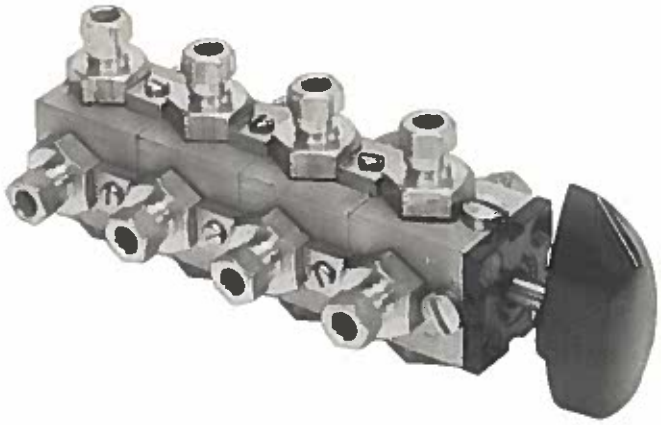


Fig 11 (A) Shuttle valve with ball element. (Air Automation).
(B) Shuttle valve allowing either one or two pressures to be applied to a single outlet without mutual interference. (CompAir Maxam).

Flow Control Valves

Flow control valves are essentially throttling devices. Their geometry can range from that of a fixed orifice to adjustable needle valves or shaped grooves, *etc*, capable of providing progressive throttling. Their basic action may also be combined with a second function such as non-return.



Drallim Award to Industry

"Yes, it's true! We award the success of our famous DRALLIM switch valve to thousands of customers who, by their application of our invention over many years to their individual switching problems, not only with air, but with liquids and gases as well, gave us the lead throughout the world with a SINGLE VALVE, with so many different configurations that it replaces the out-dated use of multi-valve conglomerates.

An Award deserves a Reward!

Let us improve your efficiency by supplying you with other valves from the same stable — Control Valves — Toggle Valves — Miniature Valves — in-line "Plan-o-line" Valves — all designed to be activated in the way you want. Better still, let us know your problems and we will provide a DRALLIM packaged solution but for your specific needs."

DRALLIM CONTROLS LTD.
BRETT DRIVE, BEXHILL-ON-SEA,
SUSSEX TN40 2JR, ENGLAND
TELEPHONE BEXHILL (0424) 216611. TELEX 95285.

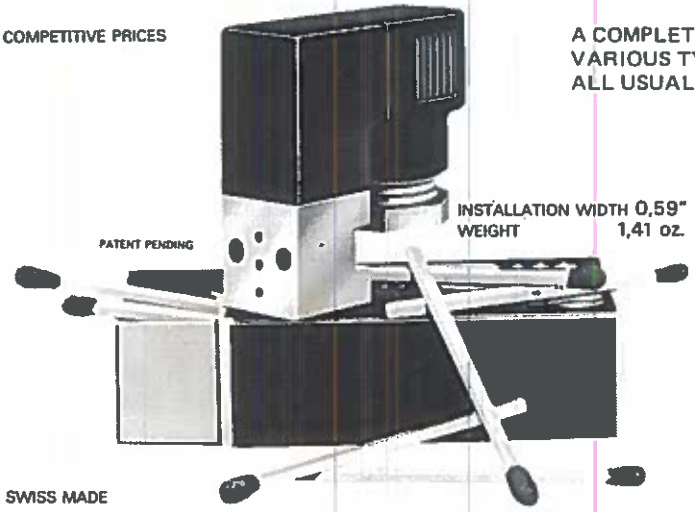
DRALLIM

MICROSOL

THE SMALLEST HIGH PERFORMANCE SOLENOID VALVE

COMPETITIVE PRICES

A COMPLETE RANGE OF
VARIOUS TYPES OF CONNECTIONS
ALL USUAL VOLTAGES



INSTALLATION WIDTH 0,59"
WEIGHT 1,41 oz.

PATENT PENDING

OF 0,5 TO 2,5 WATTS

OF 15 BARS

A LOW CONSUMPTION !!

BUT A BIG SHOT !!

SWISS MADE

FLUID AUTOMATION SYSTEMS S.A.
WE OPEN THE WAY TO PROGRESS

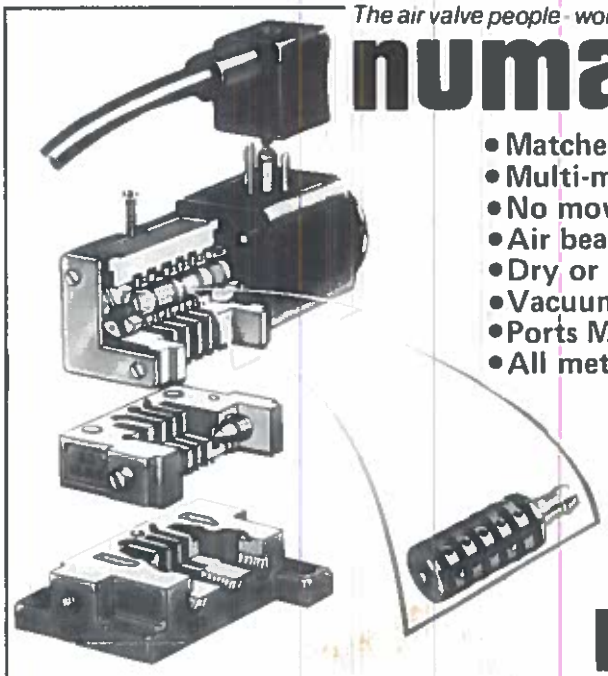
124, route de l'Etraz
CH 1290 Versoix

Tel: 55 26 91 and
55 14 93
Telex 22 194

AUSTRIA - BENELUX - DENMARK - FRANCE - FINLAND - GREAT-BRITAIN - GERMANY - HONG-KONG - ITALY - JAPAN - NORWAY - SWEDEN - SINGAPORE.

The air valve people - worldwide

numatics®



- Matched spool and sleeve assembly
- Multi-million cycle reliability
- No moving 'O' rings
- Air bearing floating action
- Dry or Lubricated service
- Vacuum to 300 PSIG
- Ports M5 to 1½ BSP
- All methods of operation available

The ultimate air valve.
Ask us to prove it —

numatics

PO BOX 18, LEIGHTON BUZZARD, BEDS. LU7 7DJ.
Telephone: 0525 370735. Telex: 826978 PNUMAT G.

Check valves, or non-return valves, are designed to provide unrestricted flow in one direction when pressurized, but close to ensure positive sealing in the absence of pressure. The common form is a suitably mounted spring loaded plunger housed in a cylindircal body seating on an internal annulus to provide a one-way seal, readily lifted off its seat by pressure on the face of the piston. The plunger end is usually fitted with an O-ring to provide the actual sealing surface. The main requirement (apart from positive sealing in the one-way direction) is that the annulus and piston should offer minimum resistance to airflow in the open position.

Exhaust Port Flow Regulators

Fig 12 shows a needle exhaust port flow regulator. This type of valve is for use in conjunction with a 5-port fully balanced spool valve for cylinder speed control. The setting of the needle regulates the rate of flow of air, to exhaust, through the control valve, thus regulating the speed of movement of the cylinder. A flow regulator is required in either of the two normal exhaust ports of a 5-port valve to control the speed of movement of the cylinder in both directions.

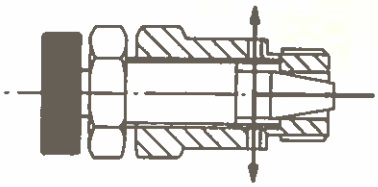
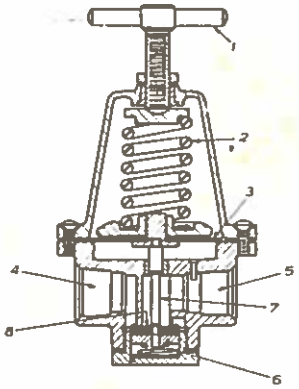


Fig 12



1 – Regulating handle.
2 – Pressure spring.
3 – Rubber diaphragm.
4 – High pressure inlet.
5 – Reduced pressure outlet.
6 – Valve return spring.
7 – Push rod.
8 – Synthetic valve washer in metal holder.

Fig 13 Reducing valve for pressures up to 7 bar (100 lb/in^2).

Pressure Control Valves

Pressure control valves may be designed to reduce the pressure to a desired level, or to maintain it at that level. A common form of construction incorporates a spring-loaded diaphragm pressurized in opposition to the spring. In the absence of pressure, the valve is held open by the spring – Fig 13. When pressure is admitted, it builds up on the outlet side until the force generated by the pressure on the diaphragm equals that of the control spring. The spring can thus be adjusted to reduce the effective outlet pressure from a maximum downwards, with reasonable accuracy of setting. In practice, the valve is simply opened enough to allow sufficient air to pass to meet the particular demand.

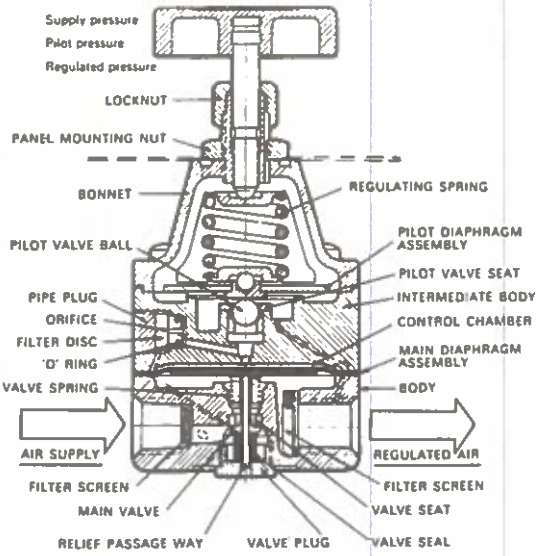


Fig 14 Precision regulating valve with pilot valve having continuous bleed controlling the pressure acting on the main valve control diaphragm. (Norgren).

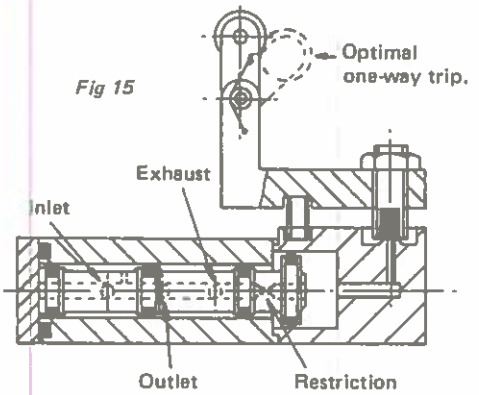


Fig 15

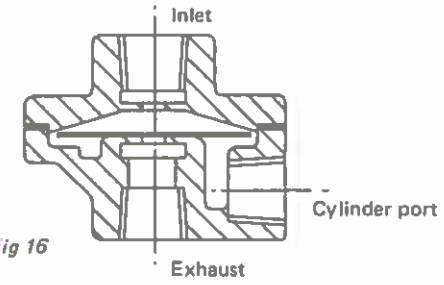


Fig 16

Fig 14 shows a more sophisticated type of pressure regulation valve which is pilot operated and has a continuous bleed off, *ie* is never completely closed. Regulation of the accuracy to ±0.5% is possible with such a device, for reasonable variations in flow and inlet pressure.

Fig 15 shows a sensitive lever controlled bleed valve. When the lever is depressed, a bleed release hole is uncovered, allowing air to escape to atmosphere and since the inflow to the pilot chamber is restricted a pressure drop occurs in the chamber and full pressure over the small end forces the piston over. Releasing the lever restores full pressure in the pilot chamber and the resultant unbalance of the end forces on the piston cause it to return to its original position. The spring loading on the lever arm is very light since it is required only to seal a 3/64" diameter orifice. A 4 oz force is sufficient to operate the valve with as little as 0.010" movement. A slightly heavier spring is fitted to the one way trip valve to ensure that the lever is not depressed in the override direction.

In application, this type of valve has the characteristics of an electrical micro-switch where short operating movements, coupled with light forces give snap action change over.

Quick Exhaust Valve

Fig 16 shows a quick exhaust valve designed to increase the speed of movement of a cylinder by allowing the exhaust air to vent directly to atmosphere. Ideally a quick exhaust valve should be screwed directly into the cylinder ports to obtain maximum effect.

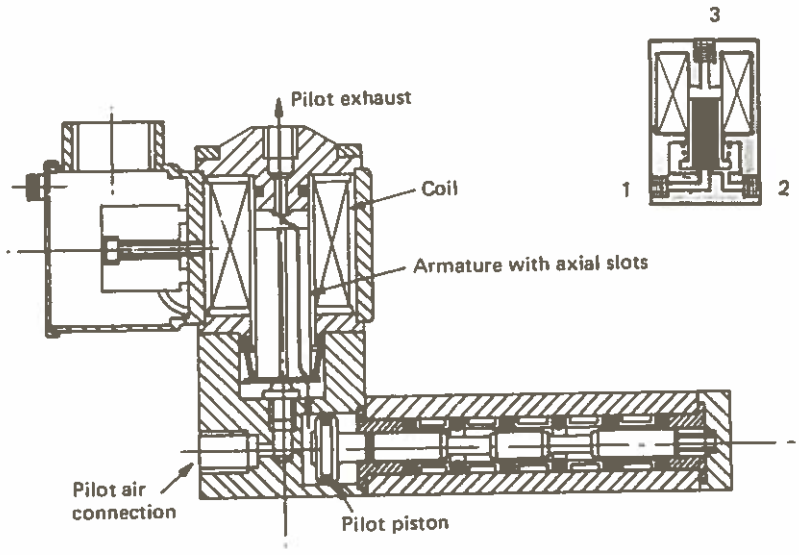


Fig 17 Three-way solenoid valve.

Solenoid Valves

When control is to be initiated by an electrical signal *(eg* a switch) some form of solenoid operator is required in order to generate the linear force necessary to operate the valve. In a simple solenoid valve this comprises an integrally mounted solenoid with the armature directly linked to the valve movement. Alternatively, to minimize the electrical power required, the solenoid may only act as a pilot control, the valve itself then being operated by fluid pressure available.

An example of this latter type of solenoid valve is shown in Fig 17. This valve controls a pilot supply for the indirect operation of the main valve spool. When the solenoid is de-energized the pilot air is blocked off and the pilot chamber is vented to exhaust by way of axial slots in the armature of the solenoid. Energizing the solenoid applies the pilot air to the pilot valve. Thus a small pilot solenoid valve operates a spool valve that would otherwise require a large solenoid with its armature directly linked to the spool. The single solenoid operated spool valve shown is normally supplied spring return, although any other mechanism could be fitted as a reset. In the same way a further solenoid could be fitted for this purpose, which would give a latching function from two independent pulses.

See also chapter on *Actuators.*

Valve Actuation

MOVEMENT OF a pneumatic valve element from one position to another may be performed by any of the following methods:

(i) *Manually* – by pushbutton, lever or pedal, *etc.*
(ii) *Mechanically* – by spring, plunger, or roller, *etc.*
(iii) *Electrically* – by solenoid (or in some cases by electric motor)
(iv) *By fluid pressure* – air-actuation.
(v) *Self-actuation.*

Configurations of these methods may also be used. Standard symbols defining these methods of actuation are given in Fig 1. (See also *Data Section* for other symbols).

From the practical point of view manual and mechanical operation can be considered basically similar in that a mechanical force is applied directly to the moving member of the valve. With lever movements the mechanical advantage can readily be adjusted. However, the particular application may demand careful consideration of the working conditions, and the possible necessity of incorporating spring detents to hold a valve in a particular position until reversed by the next movement on the part of the operator.

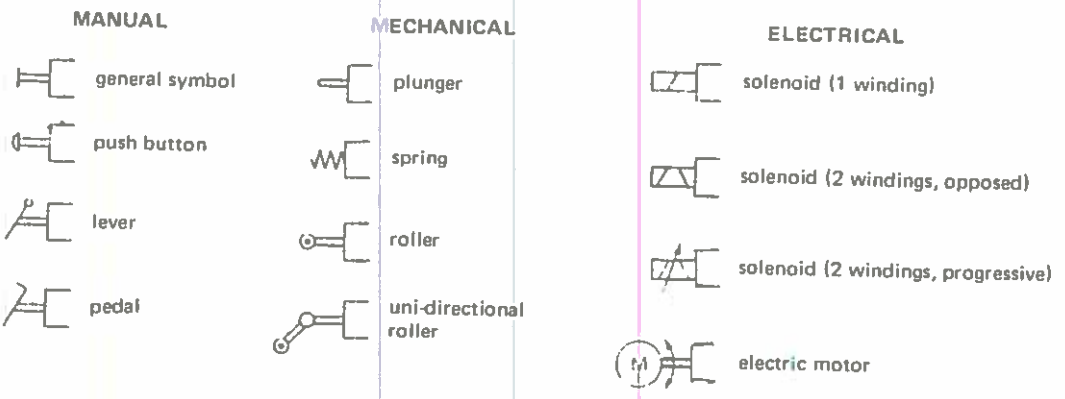


Fig 1 Standard symbols for valve control methods

Push lock button operated

Push-lever operated

Plunger operated

Palm button operated

Shrouded push button operated

Roller operated

One way trip operated

Key lock operated

Examples of manual and mechanical valve operation.

Solenoid Operation

Solenoid operation is an electro-mechanical system. The solenoid can be mounted directly on the valve, or directly connected to it by a lever, so that when an electrical switch is closed the solenoid is energized to apply direct force to the valve movement. This is *direct* solenoid actuation.

Double-solenoid actuation may be necessary with certain valves, *eg* three-position valves. If the valve is spring controlled, one solenoid is energized to shift it to one position and the second to switch it to the other position.

Double-solenoids may also be used for *momentary* working where a valve position has to be held for an appreciable time. To reduce the current drain and solenoid heating, the valve is shifted by momentary solenoid actuation, the movement being mechanically latched. The valve then remains in this position until unlatched by momentary actuation of a second solenoid releasing the catch, when it returns under spring action.

Solenoid Pilot

An alternative form is *solenoid pilot actuation.* Here the solenoid movement is utilized to open a small *pilot* valve, which then directs air pressure onto the main valve piston to initiate main valve movement. With this system a relatively small solenoid can be used to actuate quite a large valve. Another advantage is that the pilot (and its operating solenoid) does not necessarily have to be close to the main valve, but can be mounted at some remote and more convenient point to provide optimum location of both pilot and main valves. A solenoid and small pilot valve, when assembled as an integral unit, are generally referred to as *solenoid pilot.*

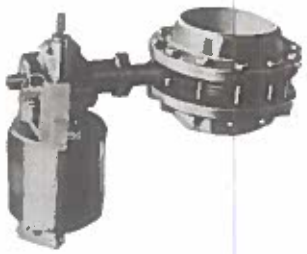
In the case of air operation this may be direct, indirect, or pilot operated. With a direct air-operated valve, signal pressure is applied directly to a piston formed by an end cap or full-bore section on the end(s) of the valve spindle. Air-operated movement can be in one direction (with spring return), or in both directions. Either method is suitable for establishing two-position movements. For three-position movements a central position can be given by applying signal pressure to both ends simultaneously, releasing pressure on alternate sides, as required.

TABLE I – ALTERNATIVE CONFIGURATIONS OF SOLENOID PILOT VALVES

Configuration	Normal Condition	Spring Action	Spring Strength	Remarks
TWO-WAY	Closed	Holding plunger against pressure force	High	Requires relatively large coil and current
	Open	Return plunger only	Light	High pressure/flow for same size coil and current
THREE-WAY	Closed	Holding only, no pressure force present	Very Light	Highest pressure/flow for same coil size and current
	Open	Return plunger only	Light	Directly comparable with 2-way, open configuration

The signal pressure required is generally substantially less than the line pressure carried by the valve. Thus lower signalling pressures can be used in many cases. However, if the signal pressure available is too low, the valve may be operated indirectly by a low pressure air signal through a suitable actuator, such as a diaphragm device. Again, one-direction or two-direction actuation is possible, by feeding signal air pressure to one or both sides of the diaphragm, respectively, with a central position obtained by pressurizing both sides of the diaphragm simultaneously.

Basically, pilot operation, or air-pilot operation, as it is more correctly described, provides the main valve with a pneumatic actuator, capable of moving the valve when the low pressure pilot signal is applied. The pilot valves used are normally three-way or four-way configuration, connecting the two chambers alternately to pressure and exhaust for pressurized operation with plug removed. Alternatively, exhaust operation would require the use of two 2-way pilot valves which allow the chambers to exhaust when opened. In this configuration the plug is removed and the chambers are permanently pressurized through the main inlet port.



Valve control by a fail-safe air operated actuator.
(Midland Pneumatic Limited).

Air Actuation

Air-actuated valves can also be designed to provide a delay in shifting the valve after receipt of pilot air ('timed-in') or cessation of the pilot supply ('timed-out'). For timed-in operation, air is admitted slowly so that a few seconds are required for enough air to be accumulated in the head to shift the valve. For timed-out operation the process is reversed, causing a delay in the exhausting of the 'pilot' air from the head. The valve thus remains actuated for a few seconds after the 'pilot' air supply is turned off. The delay mechanism can be adjusted to provide delays of varying duration. The possibility of having an automatic sequence of operations without the need for operator control is obvious with this type of valve. 'Pilot' air can be supplied automatically by some other operation of the machine and the subsequent step can be delayed for a pre-set period of time.

See also chapter on *Valves.*

SECTION 2d

Actuators

Air Cylinders

THE THREE basic types of air cylinders are shown in Fig 1, viz:

(i) *Single-acting* — or developing thrust (output force) in one direction only, with spring return.

(ii) *Double-acting* — developing thrust in both directions, pressurized from opposite sides.

(iii) *Through-rod* — usually double-acting and providing equal thrusts in either direction. (This type also provides greater stability of working since the rod is supported by two bearings.)

Variants include multiple cylinders and three-position cylinders — see later.

Parameters governing the performance of an air cylinder are:-

(a) Compressed air pressure applied to cylinder (P)

(b) Cylinder bore (D)

(c) Cylinder rod diameter (d)

The theoretical force available is:

F (outward) or +F = $0.7854 \times D^2 \times P$ — single- or double-acting cylinder

F (inwards) or $-F = 0.7854 \times (D^2 - d^2) \times P$ — double-acting cylinder.

F (inwards and outwards) = $0.7854 \times (D^2 - d^2) \times P$ — through-rod cylinder.

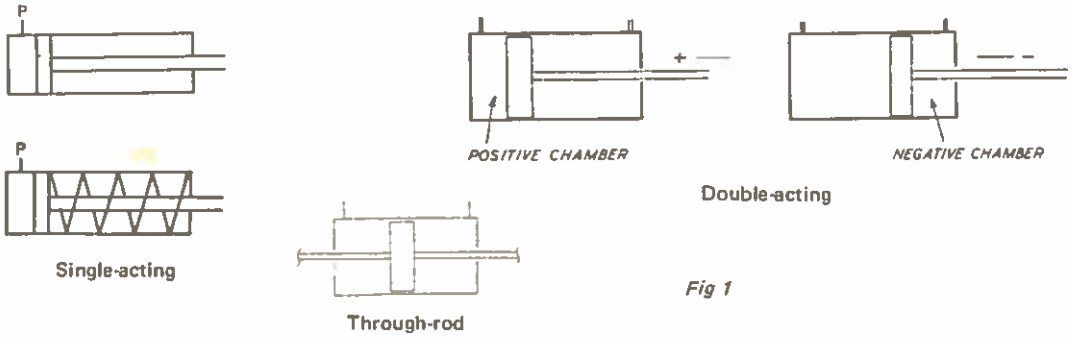


Fig 1

TABLE IA – TABLE OF FORCES FOR DOUBLE-ACTING AIR CYLINDERS – SI UNITS

Cylinder bore in mm	Piston rod direction	Maximum theoretical force in N at effective air pressures of 1–10 bar (100–1 000 kPa)									
		1	2	3	4	5	6	7	8	9	10
12	+	11	22	34	45	56	68	79	90	101	113
	–	10	21	32	42	51	62	72	82	92	103
16	+	20	40	60	80	100	140	160	180	200	
	–	19	36	52	70	87	105	122	140	168	175
20	+	31	62	94	126	154	188	220	251	285	314
	–	29	51	85	114	140	170	199	237	258	284
25	+	49	98	147	196	245	294	343	392	441	490
	–	38	76	114	152	190	228	266	304	342	380
32	+	80	160	240	320	400	480	560	640	720	800
	–	69	138	207	276	345	414	483	552	621	690
40	+	126	252	378	504	630	756	882	1 008	1 134	1 260
	–	106	212	318	424	530	636	742	848	954	1 060
50	+	200	390	590	780	980	1 180	1 370	1 570	1 760	1 960
	–	170	330	500	660	830	990	1 160	1 320	1 490	1 650
63	+	310	620	940	1 250	1 560	1 870	2 180	2 500	2 810	3 120
	–	280	560	840	1 120	1 400	1 680	1 960	2 240	2 520	2 300
80	+	500	1 000	1 510	2 010	2 510	3 010	3 510	4 020	4 520	5 020
	–	450	910	1 360	1 810	2 270	2 720	3 170	3 620	4 080	4 530
100	+	790	1 570	2 360	3 140	3 930	4 710	5 500	6 280	7 070	7 850
	–	710	1 410	2 220	2 820	3 530	4 230	4 940	5 640	6 350	7 050
125	+	1 230	2 450	3 680	4 910	6 140	7 360	8 590	9 820	11 040	12 270
	–	1 150	2 290	3 440	4 580	5 730	6 880	8 020	9 120	10 310	11 460
150	+	1 760	3 520	5 280	7 040	8 800	10 560	12 320	14 080	15 840	17 600
	–	1 660	3 320	4 970	6 730	8 300	9 950	11 620	13 250	14 920	16 600
160	+	2 010	4 020	6 030	8 040	10 050	12 060	14 070	16 080	18 090	20 100
	–	1 890	3 770	5 650	7 540	9 420	11 300	13 190	15 070	16 960	18 850
200	+	3 140	6 280	9 430	2 570	15 710	18 850	21 990	25 140	28 280	31 420
	–	3 020	6 030	9 050	2 060	15 080	18 100	21 110	24 130	27 140	30 160
250	+	4 920	9 840	14 760	18 550	24 600	29 520	34 340	39 360	44 280	49 200
	–	4 640	9 270	13 900	19 680	23 200	27 800	32 500	37 100	41 750	46 400
300	+	7 080	14 100	21 240	28 320	35 400	42 480	49 560	56 640	63 720	70 800
	–	6 700	13 400	20 100	26 800	33 500	40 200	47 000	53 700	60 200	67 000
350	+	9 620	19 240	28 860	38 480	48 100	57 720	67 340	76 960	86 580	96 200
	–	9 120	18 200	27 300	36 500	45 500	54 700	63 700	72 800	82 000	91 200
400	+	12 580	25 160	37 740	50 320	62 900	75 480	88 060	100 640	113 220	125 800
	–	11 800	23 600	35 400	47 200	59 000	70 080	82 750	94 500	106 300	118 000
450	+	15 900	31 800	47 700	63 600	79 500	95 400	111 300	127 200	143 100	159 000
	–	14 950	29 000	44 800	59 750	74 750	89 600	104 500	118 500	134 200	149 500
500	+	19 650	38 300	58 950	78 600	98 250	117 900	137 550	157 200	177 750	196 500
	–	18 250	37 000	55 600	74 200	92 700	112 000	130 500	143 200	167 800	185 200

1 N (Newton) = 0.1 kp (kilopond)
1 bar = 100 kPa
The values in the table give the forces for the plus and minus movements at air pressures of 1–10 bar (100–1 000 kPa)

Force figures for cylinders are normally quoted as full theoretical value, called the *static thrust rating* – see Table I. This does not necessarily give an accurate estimate of working force available in a particular application. Two additional factors have to be considered here.

(i) Actual pressure developed in the cylinder chamber, which will be less than the nominal line pressure.

(ii) Speed of operation. With slow speeds, working thrusts are likely to be 85% of the theoretical value for small cylinders and 90% of the theoretical value for large cylinders. At higher speeds, working thrusts are still further reduced (*eg* down to 50% of the theoretical value in some cases).

Note, too, that with single-acting spring-return cylinders the thrust is reduced by the amount of spring force overcome on the working stroke.

The only real value of a static thrust figure is to indicate the likely maximum 'break out' force available from an air cylinder. As soon as stroke motion is induced the thrust force will be reduced by friction (mechanical loss) and compressed air pressure drop through the inlet port (reduction in effective pressure in the operating chamber).

TABLE IB – FORCES FOR DOUBLE-ACTING AIR CYLINDERS – US UNITS

Cylinder bore	Piston rod direction	Maximum theoretical force in pound-force (lbf) at effective air pressures of 20–140 lb/in^2												
In		20	30	40	50	60	70	80	90	100	110	120	130	140
0.47	+	3.5	5.3	7.0	8.8	10.5	12.3	14.0	15.8	17.5	19.3	21.0	22.8	24.5
	–	3.1	4.7	6.2	7.8	9.4	10.9	12.5	14.0	15.6	17.2	18.7	20.3	21.8
0.63	+	6.2	9.4	12.5	15.6	18.7	21.8	25.0	28.1	31.2	34.3	37.4	40.6	43.7
	–	5.4	8.0	10.7	13.4	16.1	18.8	21.4	24.1	26.8	29.5	32.2	34.8	37.7
0.79	+	9.8	14.6	19.5	24.4	29.3	34.2	39.0	43.9	48.8	53.7	58.6	63.4	68.3
	–	7.8	11.7	15.6	19.6	23.5	27.4	31.3	35.2	39.1	43.0	46.9	50.8	54.7
0.98	+	15.2	22.8	30.4	38.0	45.6	53.2	60.8	68.4	76.0	83.6	91.2	98.8	106.4
	–	11.7	17.6	23.4	29.3	35.1	40.9	46.8	52.6	58.5	64.3	70.2	76.0	81.9
1.26	+	24.9	37.4	49.8	62.3	74.8	87.2	99.7	112.1	124.6	137.1	149.5	162.0	174.4
	–	21.4	32.2	42.8	53.5	64.3	74.9	85.7	96.3	107.1	117.1	128.5	139.2	149.9
1.57	+	38.9	58.4	77.8	97.3	116.7	136.2	155.6	175.1	194.5	214.0	233.4	252.9	272.3
	–	37.2	49.1	65.4	81.7	98.0	114.4	130.7	147.1	163.4	179.8	196.1	212.5	228.8
1.97	+	60.8	91.2	122	152	182	213	243	274	304	334	365	395	426
	–	51.1	76.6	102	128	153	179	204	230	255	281	306	332	357
2.48	+	96.6	145	193	242	290	338	386	435	483	531	580	628	676
	–	86.9	130	174	217	261	304	347	391	434	478	521	565	608
3.15	+	156	234	312	390	467	545	623	701	779	857	935	1 013	1 091
	–	141	211	281	351	422	492	562	633	703	773	844	914	984
3.94	+	244	366	488	610	732	854	976	1 098	1 220	1 844	1 466	1 588	1 710
	–	219	329	438	548	657	767	876	986	1 095	1 205	1 815	1 424	1 534
4.92	+	380	570	760	950	1 140	1 330	1 520	1 710	1 900	2 090	2 280	2 470	2 660
	–	355	533	710	888	1 065	1 243	1 420	1 598	1 775	1 953	2 131	2 308	2 486
6.30	+	624	936	1 248	1 560	1 872	2 184	2 496	2 808	3 120	3 432	3 744	4 056	4 368
	–	585	878	1 170	1 463	1 755	2 048	2 340	2 633	2 926	3 218	3 511	3 803	4 006
7.87	+	975	1 462	1 948	2 430	2 924	3 410	3 896	4 378	4 860	5 354	5 848	6 334	6 820
	–	912.5	1 370	1 825	2 290	2 740	3 195	3 650	4 115	4 580	5 030	5 480	5 935	6 390
9.84	+	1 525	2 290	3 060	3 805	4 580	5 350	6 120	6 865	7 610	8 385	9 160	9 930	10 700
	–	1 438	2 155	2 880	3 530	4 310	5 035	5 760	6 410	7 060	7 840	8 620	9 345	10 070
11.81	+	2 195	3 310	4 400	5 480	6 620	7 710	8 800	9 880	10 960	12 100	13 240	14 330	15 420
	–	2 075	3 120	4 170	5 280	6 240	7 290	8 340	9 460	10 580	11 530	12 480	13 530	14 580
13.78	+	2 980	4 475	5 970	7 440	8 950	10 445	11 940	13 410	14 880	16 390	17 900	19 395	20 890
	–	2 825	4 230	5 660	7 075	8 460	9 890	11 320	12 735	14 150	15 535	16 920	18 350	19 780
15.75	+	3 870	5 840	7 820	9 760	11 680	13 660	15 640	17 580	19 520	21 440	23 360	25 340	27 320
	–	3 655	5 480	7 330	9 150	10 960	12 810	14 660	16 480	18 300	20 110	21 920	23 770	25 620
17.72	+	4 925	7 380	9 850	12 340	14 760	17 230	19 700	22 190	24 680	27 100	29 520	31 990	34 460
	–	4 630	6 740	9 275	11 580	13 480	16 015	18 550	20 855	23 160	25 060	26 960	29 495	32 030
19.69	+	6 080	9 130	12 200	15 240	18 260	21 330	24 400	27 440	30 480	33 500	36 520	39 590	42 660
	–	5 740	8 610	11 500	14 480	17 220	20 110	23 000	25 980	28 960	31 700	34 440	37 330	40 220

The values in the table give the forces for the plus and minus movements at air pressures of 20–140 lb/in^2

This can present problems in cylinder size-selection, since the thrust produced by a given size of cylinder can be modified by the use to which it is put (and particularly the speed of working). For non-critical applications, air cylinders are commonly sized on the basis of a theoretical thrust rating of 25% to 50% in excess of the working force required. It is better to select an oversized cylinder rather than the reverse, since the thrust provided by an 'undersized' cylinder can only be uprated by increasing the line pressure (which is seldom a practical solution), or by reducing the operating speed.

In cases of uncertainty, a solution is to select a suitable size of cylinder with the performance assessed on a lower figure than the pressure available. The cylinder is then fed through a pressure reducing valve. If necessary, the cylinder force can then be increased via the pressure control valve.

For more critical applications it is generally necessary to consider in detail:

(i) actual power requirements, taking into account losses due to positive and negative accelerations.

(ii) any additional power requirements which may be necessary to ensure constant motions under varying load conditions.

(iii) how to calculate the actual value of the effective pressure under working conditions.

The majority of air cylinders are designed for working with maximum air pressures of the order of 10 bar (140 lb/in^2), although the usual operating pressure is of the order of 6–7 bar (80–100 lb/in^2) which is consistent with the typical delivery air pressure available from the bulk of standard production compressors.

Air Consumption

The air consumption of a cylinder can be derived directly from the cylinder geometry. It is normally expressed in terms of free air delivered.

+ stroke $$m^3/\text{min (FAD)} = \frac{0.7854\ D^2 \times S \times SPM \times (P + 1)}{10^9}$$

where S and D are in millimetres
P = pressure in bar
SPM = stroke per min

+ stroke $$ft^3/\text{min (FAD)} = \frac{0.7854\ D^2 \times S \times SPM \times (P + 14.7)}{1728 \times 14.7}$$

where S and D are in inches
P = pressure in lb/in^2 gauge

– stroke $$m^3/\text{min (FAD)} = \frac{0.7854\ (D^2 - d^2) \times S \times SPM \times (P + 1)}{10^9}$$

where d = rod diameter in millimetres

– stroke $$ft^3/\text{min (FAD)} = \frac{0.7854\ (D^2 - d^2) \times S \times SPM \times (P + 14.7)}{1728 \times 14.7}$$

where d = rod diameter in inches

Double-acting cable cylinder.

In the case of double-acting cylinders the reduction in air consumption on the –stroke is commonly ignored and taken as twice that of the +stroke.

Total consumption is calculated by multiplying the consumption per stroke by the number of strokes per minute. It is often more convenient to render consumption directly in terms of the operating time to complete a stroke, when the corresponding formulas are:

$$m^3/min\ (FAD) = \frac{D^2 \times S \times (P + 1)}{t \times 10^4}$$

$$ft^3/min\ (FAD) = \frac{0.00186\ D^2 \times S \times (P + 14.7)}{t}$$

where t = operating time in seconds

Time of Operation

The operating time of an air cylinder is controlled partly by load inertia and partly by the size of the inlet port. The latter produces a 'throttling' effect, with pressure drop increasing as the piston speed (and hence air delivery through the port) increases. The basic relationship between consumption and piston speed is:

$$Q = D^2 \times V \times r$$

where V = piston speed
r = compression ratio
Q = consumption (FAD)

Without throttling, piston velocity is finally restricted by the pressure drop through the inlet port. With small inlet ports and light external loads, pressure drops of the order of 2–3 bar (30–40 lb/in^2) may readily be realized, resulting in a halving of the output force available from normal line pressures. Individual cylinders will, therefore, have a characteristic speed performance, determined by their bore size and port size.

The fitting of a silencer to the exhaust port of a cylinder control valve can also have a speed control effect. This will be dependent on the size and type of silencer, *ie* its effective permeability. In practice the effect of a suitable size of silencer capable of giving a significant reduction in exhaust noise on the piston can vary from zero to a maximum of about 20% loss of speed.

Normally the speed range likely to be available from a cylinder is 0.5–2.5 metres/sec (1.6–8 ft/sec). Attempting to operate at lower or higher speeds can produce difficulties.

Speed Control

The speed of operation of an air cylinder can be controlled by throttling either the inlet or outlet air or both. The degree of speed control available from such simple methods is suitable for a wide

range of applications although the actual response time will still be variable if the load is variable, because of the varying compressibility of air against different loads. Actual control of speed as such, however, is often less important than reducing the speed of operation to a satisfactory level or avoiding excessive operating speeds which increase wear on the cylinder and waste air.

Speed control accomplished by metering the inlet air will only produce a constant speed if the resisting force is constant – Fig 2. Basically, pressure will first build up in the cylinder until the force available is greater than that of 'stiction' (static friction) and resistive load, when the piston will start to move. It will then continue to move forward at a more or less constant speed if the resistive load is constant, with speed governed by the rate of air admission. Any increase or decrease in resistive load will cause the speed of movement to slow down or increase respectively.

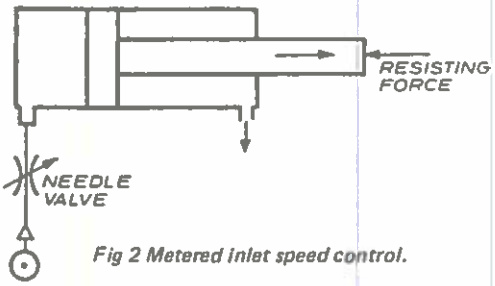


Fig 2 Metered inlet speed control.

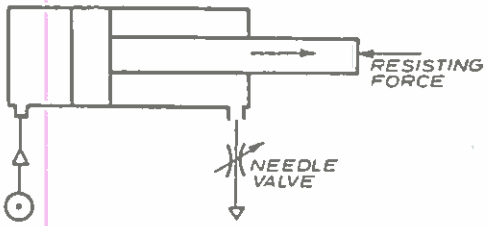


Fig 3 Metered exhaust speed control.

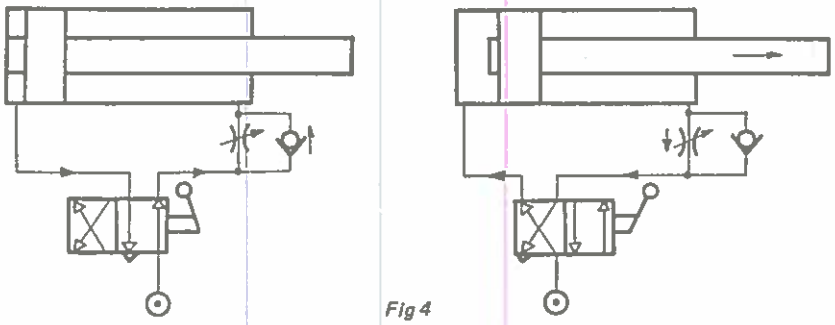


Fig 4

Speed control by metered exhaust – Fig 3 – will tend to give an initial rapid movement followed by a slowing down as air is compressed on the exhaust side of the piston. The movement is thus likely to be jerky. Much better control is provided if the modified circuit of Fig 4 is used where the piston is held in the retracted position under pressure and, when the control valve is operated, the piston will start to move forward through pressure applied to the head before the air has had time to leak away from the rod end. The result is a freedom from initial jerk and smooth movement throughout. Also the tendency to jerk forward should the resisting force be removed is minimized.

The general recommendation is that speed control should preferably be applied by throttling the exhaust from a cylinder. A combination of both methods may, however, be used in specific applications.

For more precise speed control, and particularly to take care of fluctuations in load and/or line pressure, surge damping may be applied to the system. This would normally demand the use of an 'oversize' system with a surge tank in the circuit, forming the equivalent of an accumulator in a hydraulic circuit. For greater accuracy of control, however, a check unit would be employed.

Check Units

A typical check unit consists of a small hydraulic cylinder with both ends connected to a suitable size of reservoir. The hydraulic cylinder is mechanically connected to the air cylinder so that flow in and out of the hydraulic cylinder (displaced by its piston movement) is a function of the air cylinder movement. By inserting regulating valves in the hydraulic cylinder circuit the movement is checked or regulated, thus regulating the speed of movement of the whole assembly. Control and rigidity of the system are then directly comparable with those of an all-hydraulic system.

The hydraulic cylinder can be a separate entity (with mechanical linkage) or built integral with the air cylinder in a common unit, more usually known as an air-hydraulic cylinder. Speed of movement is then controlled by a metering valve in the hydraulic circuit whilst variations in oil volume through temperature changes or leakage are catered for by the reservoir — Fig 5.

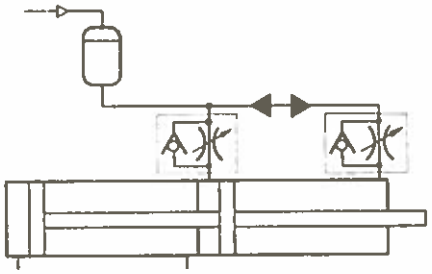


Fig 5 Hydro check unit using an air-hydraulic cylinder.

Further variations are possible, such as bypassing the (hydraulic) control valve for a proportion of the movement to give a rapid initial movement followed by checking and positive control. It should be noted, however, that the controlled speed as established by a particular setting of the (hydraulic) metering valve will still be influenced by the actual value of the resisting load, although speed variations with a fluctuating load will be greatly smoothed. In a more sophisticated system it is possible to couple the metering adjustment to the actual resistive load to make the system fully self-compensating and so maintain constant speed. For most practical applications adequate precision in speed control can usually be obtained in such systems by employing an oversize cylinder and so rendering it less subject to load fluctuation effect, when the proportionately smaller variations can be dealt with adequately by the check unit.

With light resistive loads, and without check units, the capability of air cylinders to perform rapid stroking motions can also be a disadvantage in that considerable kinetic energy may be dissipated at the end of the stroke. This can set up severe stresses in the cylinder itself or its end covers, or on the frame of the machine to which the cylinder is attached. This can be overcome by providing cushioning to produce gradual deceleration of the piston as it approaches the end of its stroke. Thus, in general, cushioned cylinders are to be preferred to reduce shock and 'hammer' at the end of the operating stroke, unless there are obvious reasons for employing a non-cushioned type.

Cushioning

Since air cylinders are fast-acting and the piston will tend to accelerate over the whole of the stroke (unless restricted by throttling or load), considerable shock loads can be developed if the

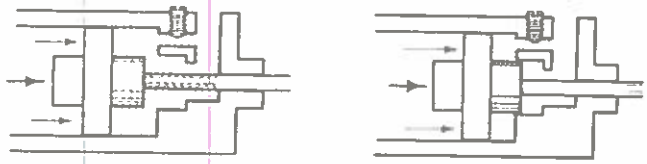


Fig 6A Principal of air cylinder cushion. (CompAir Maxam Ltd).

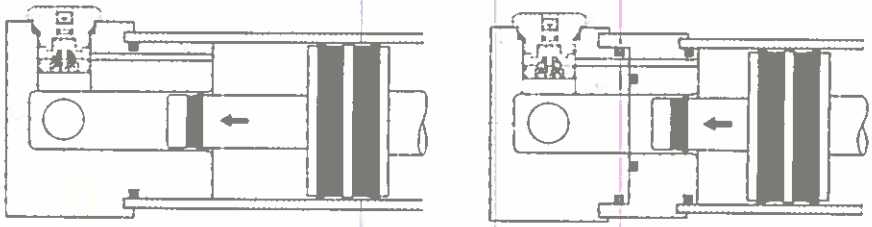


Fig 6B Extended cushions for cylinders. (Mecman).

piston is allowed to impact on the end of the cylinder. To avoid this a decelerating device or *cushion* is commonly incorporated in the cylinder geometry. This normally takes the form of a cushion chamber incorporated in the end of the cylinder, into which a nose section on the piston enters – Fig 6.

The effect is that as the piston approaches the end of its stroke and the nose section enters the chamber, air is trapped in this chamber, from which it can only escape through a controlled bleed. This provides adjustable damping or 'braking' over the length of the cushion travel, with the kinetic energy of the piston mass dissipated in the form of heat.

Provision must also be made for reverse air feed to act on the full area of the piston, otherwise initial acceleration and break-out force available will be very low. A common arrangement is to fit a non-return valve in the end of the cover so that reverse flow is directed straight to the face of the piston, the end cover or piston having some form of fluting to ensure that pressurized air is distributed over the whole face of the piston.

Detail design of cushions varies with different manufacturers. The cushion length is usually only a matter of 25 mm (1 inch) or so in most cylinders. The cushion boss or piston nose is usually tapered to provide a smooth, progressive entry, but provides a plunger-type seal once it has fully entered the cushion chamber. Alternatively, ring seals may be used in place of plunger-type seals.

A cushion can be fitted to one or both ends of a cylinder. The cushion chamber may be incorporated in the end cover(s) or be in the form of a cushion collar.

Variations in integral cushion design include the use of a spring-loaded face seal which forms its own cushion automatically when brought into contact with a seating surface at the end of the cylinder; and flexible seal type cushions. In the latter case the seal is made flexible enough to collapse when return flow is applied, thus opening the full area of the piston to the flow. Other types, incorporating a sliding plate valve, may permit adjustment of the cushion length to meet different working conditions.

Integral cushions have certain distinct limitations, the main being that at speeds above about 4.5–5.5 metres/second (15–18 feet/second) they are virtually ineffective, unless the cushion length is made impractically long. For cushioning higher piston speeds, external cushioning is normally required. This can take the form of flow control where restriction is applied to the

exhaust flow over a selected portion of the stroke, starting at a point well before that practicable with integral cushions. This normally involves the use of a pilot valve, suitably positioned for operating by the cylinder movement, controlling a diverter valve which applies restriction to the exhaust flow. If necessary, this set up can be duplicated, or triplicated, *etc*, to provide a number of different stages of cushioning.

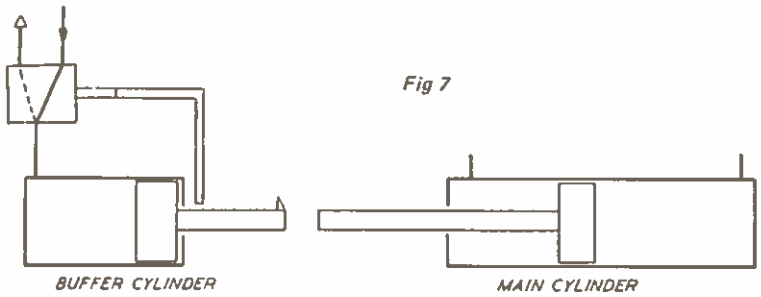


Fig 7

Buffer Cylinder

Instead of incorporating an integral cushion a buffer cylinder may be used as decleration control. This is a separate cylinder pre-pressurized and proportioned to absorb the excess momentum — Fig 7. The main requirement is that the buffer cylinder should have reserve capacity, and that the maximum pressure realized in the buffer cylinder should not exceed the design rating (which can be very much higher than that for conventional air cylinders). With suitable design, cushioning of high speeds and forces can be provided by quite short buffer cylinders pre-pressurized to the order of 0.7 to 1 bar (10 to 15 lb/in^2).

The valving circuit associated with a buffer cylinder must be capable of reversing the main control valve at the end of the buffer stroke, otherwise the high pressure generated in the buffer cylinder will tend to reverse the travel of the load.

Other limitations of integral cushions include the fact that they are generally ineffective where a cylinder is fitted with quick exhaust valves or is pre-exhausted, as the amount of air trapped in the cushion is likely to be inadequate for proper cushioning; there is also the practical problem that the bleed may be subject to clogging by contaminants unless periodically attended to. Further of course, integral cushions are only effective if the full stroke of the cylinder is used.

Nevertheless cylinders with integral cushions have a wide application and are generally capable of giving excellent performance in the majority of systems. Piston speed limitation is less likely to be a factor than may appear at first sight, since air cylinders are seldom required to operate at their fastest speed, and the cylinder operating speed is frequently controlled.

Positioning Cylinders

A *positioning cylinder* is one designed to provide intermediate movements or positioning of the piston. It is normally produced by adding an external control and/or indicating device known as a *position controller.* A position controller provides a means of measuring or determining the position of the piston in the cylinder. Within limits, the position is largely unrelated to the load on the cylinder, with the position controller itself controlled by a variable pressure air supply. This can either be varied manually, or automatically by a suitable instrument or control gear, depending on the application involved. The most common application is where a quantity has to be varied continually, such as in air-operated valves, speed variators and similar devices.

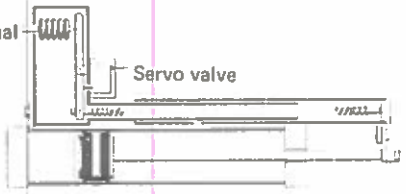


Fig 8 Principle of operation of a positioning cylinder. (Atlas Copco).

Individual cylinders may also be adapted for mechanical or pneumatic signalling and control, governed purely by piston movement. The simplest method is to employ limit switches or strikers fitted externally and operated by the rod movement (Fig 8).

Adjustable Stroke Cylinders

For certain applications, a cylinder with an adjustable stroke may be required. In its simplest form this can comprise a standard double-rod cylinder with one rod end threaded where it emerges from the cylinder. Check nuts are then run on to this thread to limit the available movement, and thus the stroke of the cylinder, to that required. In such designs it is important that the threads are not allowed to run into the end gland as otherwise the seal will be damaged resulting in air leakage. This can be avoided by extending the rod so that the threaded length never enters the gland, and fitting a sleeve over the rod extension (or an extended housing to the cylinder end) to act as a stop for the check nuts.

Various other methods of stroke adjustment can be used, according to the circumstances and requirements, ranging from adjustable external stops, which are not part of the cylinder as such (and are thus applicable to standard cylinders) to piston travel limit via a large stud screwed into the end cover and locked with a check nut. In such cases the stud must be of sufficient size and strength to absorb the continual impact of the piston without damage to either component; in the case of double-acting cylinders it will be necessary to provide a seal for the stud thread.

Tandem Cylinders

Tandem cylinders are produced in two different configurations — separate cylinders with the piston mounted on a common rod, and a common cylinder with separate pistons, back-to-back. These are known as *front connected* and *rear connected* respectively — Fig 9.

Tandem cylinders may be used where there is insufficient space available to install standard cylinders of the appropriate diameter necessary for the force required. Control is generally similar to that of a conventional cylinder, except that compressed air is fed simultaneously to the two cylinder chambers.

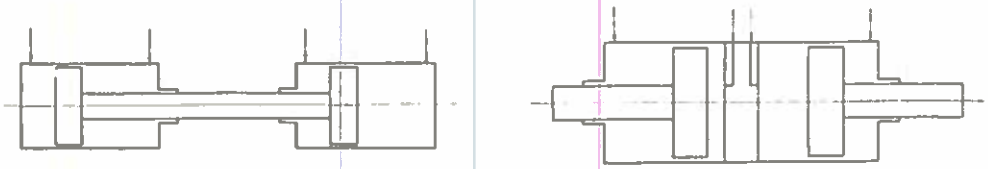


Fig 9 Tandem cylinders — front connected (left) and rear connected (right).

Three-Position Cylinders

Three-position cylinders can be used for operating mechanisms which require three fixed stop positions — *ie* one at an intermediate position as well as a stop position at each end. The basic

configuration is shown in Fig 10. This is really another form of tandem cylinder, except that the two piston assemblies can move independently. The blind end cylinder (referred to as the 'stop' cylinder) provides the intermediate stop position on its + stroke, and again limits the – stroke movement of the main piston if held in this position. The two cylinders may be of equal, or different, diameters, depending on the power ratios required. See also Fig 11.

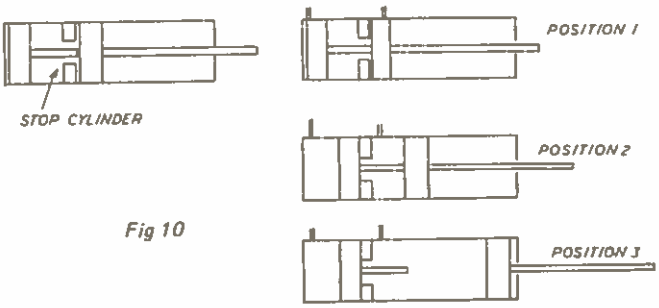


Fig 10

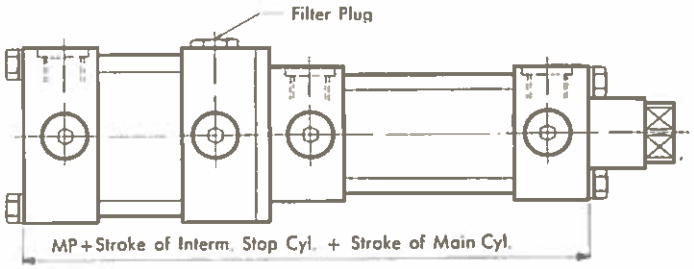


Fig 11 Three-position cylinder. (Mecman).

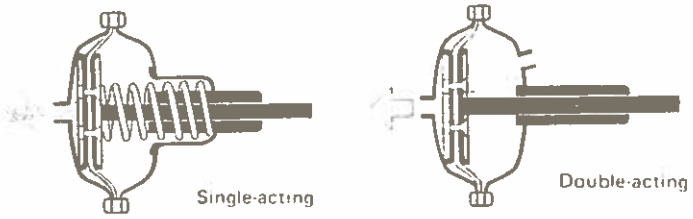


Fig 12 Single- and double-acting diaphragm cylinders. (Atlas Copco).

Thrusters (Diaphragm Cylinders)

A diaphragm cylinder is basically a large diameter, short stroke cylinder fitted with a diaphragm instead of a piston. Construction normally takes the form of two shallow steel pressings or castings clamped or bolted together with the diaphragm. Such cylinders may be single-acting or double-acting – see Fig 12.

Diaphragm cylinders are capable of generating very high forces with very short strokes and are commonly called *thrusters*. Theoretical thrust available is equal to the product of the applied air pressure and effective diaphragm area. Stroke is normally limited to a maximum of about one third the cylinder diameter. Normal mounting position is with the rod horizontal or pointing downwards.

Impact Cylinders

An 'impact' cylinder is one arranged so that the speed developed is high enough for impact work, such as forging, piercing, *etc.*

It is possible to achieve this mode of working with a conventional cylinder fitted with extra large inlet and outlet ports and suitable valving. Such a system is shown in Fig 13. This provides for the air on the underside of the piston to be quickly exhausted, while simultaneously supplying air to the other side. The piston speed obtained can be varied by adjusting the appropriate valves. It is generally advisable — or even necessary — to use a reservoir of at least equivalent capacity to that of the cylinder displacement and which is connected close to the cylinder in order to avoid too high a pressure drop.

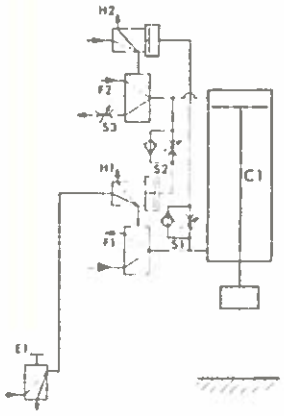
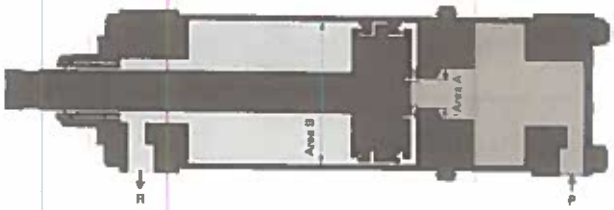


Fig 13 Circuit for operating an air cylinder as an impact cylinder. (Mecman).



Section through impact cylinder. Piston remains at rear end position (seat closed on pre-pressurizing chamber) until product of inlet pressure P times area A is greater than product of outlet pressure R times area B. Exhaust rate controlled by downstream throttle valve.

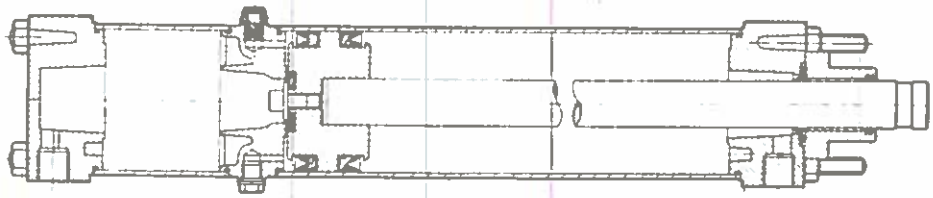


Fig 14 Martonair impact cylinder.

An alternative approach is to incorporate a reservoir section in the cylinder, connecting directly to a large inlet port valve. Such a design is shown in Fig 14. Initially the piston rod end is charged with air at normal line pressure with the annulus between the piston seat and centre piece connected to exhaust. The control valve is then opened and pressure builds up in the reservoir. This acts on the piston seat, which is 1/n that of the piston area (*ie* A/n where n is the ratio of the piston to annulus area). The thrust force downwards is thus P x A/n.

The piston rod end of the cylinder is connected to exhaust when the pressure in this end falls to less than the supply pressure P. When it has fallen to just below P/n the piston moves away from

the poppet seat. Immediately this happens the whole of the piston is exposed to the reservoir pressure causing the piston to move rapidly downwards. Maximum piston speed is reached within the first 25–35 mm of the stroke and is of the order of 8 metres/sec (25 ft/sec) with supply pressure of 6 bar.

Because of the high shock loads developed, a necessary requirement for all impact cylinders is that they be mounted rigidly. For even higher energy outputs pairs of cylinders can be mounted in opposition and operated simultaneously. With proper alignment, shock loading can be negligible in such cases, all the energy being absorbed by the workpiece. Some impact cylinders are fitted with a cushion to prevent damage if they are used for other types of work.

Linear Motors

A double-acting cylinder can be made to perform continuous stroking out and in by fitting trip valves or similar devices which automatically remove the supply of compressed air at the end of each stroke — Fig 15. The motion of such a cylinder is then continuous reciprocation, and in this mode of operation it is called a *linear motor.*

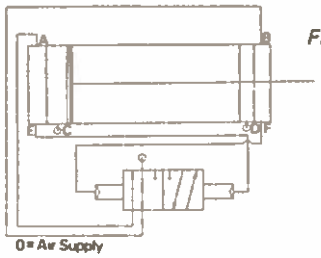
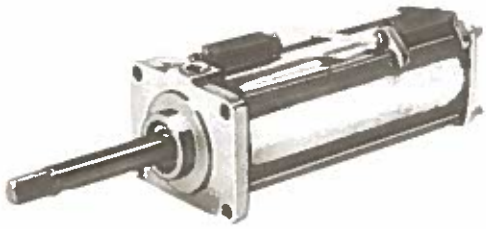


Fig 15 Circuit for air cylinder operating as a linear motor. (Mecman).



'Pneumag' cylinder with magnetic proximity switches mounted on tie-bars.

Materials and Construction

Air cylinders are generally produced in three different grades or specifications, namely *light duty, medium duty* and *heavy duty.* Such a classification is based on the rigidity of cylinder construction and its suitability for accommodating working loads normally associated with these three grades. It does not refer to the output performance of the cylinder, which will be similar for cylinders of similar bore size operating at the same working pressure.

Air cylinders are most commonly assembled from hard drawn metallic cylinder tubes with fabricated end covers. The rod end cover is extended to incorporate a bearing for the piston rod and rod seals. The end covers also normally incorporate the cylinder ports and cushion chambers where fitted. Choice of constructional materials varies both with the duty and size of cylinder — *eg* see Table II.

Light duty cylinders:

Tube-plastic or hard drawn aluminium or brass.
End covers — cast aluminium or machined from aluminium or brass.
Piston — machined from aluminium, or cast aluminium.
Piston rod — mild steel (chrome plated preferred).

TABLE II – TYPICAL STANDARD SIZE CYLINDER GEOMETRY

Inch Size Cylinders

Cylinder Diameter in	Rod Diameter in	Typical Port Size
¾	5/16	BSP 1/8 in
1	3/8	BSP 1/8 in
1½	½ or 5/8	BSP 1/8 in
2	5/8 or ¾	BSP ¼ in
2½	¾	BSP ¼ in
3	1	BSP 3/8 in
4	1¼	BSP 3/8 in
5	1¼ or 1½	BSP 3/8 in
6	1.3/8, 1½ or 1¾	BSP ½ in
8	1½ or 1¾	BSP ¾ in
10	1¾, 2 or 2¼	BSP 1 in
12	2¼, 2½ or 3	BSP 1¼ in
14	2½ or 3	BSP 1½ in

Metric Size Cylinders

Cylinder Diameter mm	Rod Diameter mm	Typical Port Size
12	4	M5
16	6	
20	8	BSP 1/8 in
25	12	or M5
32	12	
40	16	
50	20	BSP ¼ in
63	20	BSP ¼ in
80	25	BSP 3/8 in
100	32	BSP 3/8 in
125	32	BSP 3/8 in
150	35	BSP ½ in
200	50	BSP ¾ in
250	60	BSP 1 in
300	70	BSP 1¼ in
350	80	BSP 1.1/8 in
400	100	BSP 1½ in
450	110	BSP 2 in
500	120	BSP 2 in

Plastic cylinders may be all-plastic except for the piston rod, or mixed metal and plastic. For example, a plastic cylinder tube and end covers with a metal piston and stainless steel rod may have advantages when working in a corrosive ambience. Plastic materials used include nylon, polyacetal, polycarbonate, rigid PVC and reinforced epoxy resins.

Apart from corrosion resistance, plastic cylinders can also offer cost advantages. However the higher moduli associated with plastic materials can restrict the working temperature range of plastic cylinders and substantially reduce the maximum service temperature rating.

Medium duty cylinders:

Tube — hard drawn brass or cast aluminium.
End covers — machined or cast aluminium, brass or bronze; or cast steel.
Piston — machined or cast aluminium, brass, bronze or steel.
Piston rod — steel chrome plated steel, hardened steel or stainless steel.

Heavy duty cylinders:

Tube — hard drawn brass or steel; cast brass, bronze, cast iron or steel; welded steel tube (large cylinders only).
End covers — high tensile castings.
Piston — cast aluminium, brass, bronze, iron or steel; machined brass or bronze; aluminium forging.
Piston rod — chrome-plated steel, hardened steel, stainless steel.

Various methods are used for attaching the end covers, including screwing, welding and mechanically fastening (*eg* screwing into a square cylinder body). Tie-rod assembly is the most widely adopted method, enabling substantial end covers to be used, square in shape, with the assembly bolted up with the tie-rods in tension. These rods are normally of high tensile steel. Pre-stressing of the tie-rods by tightening up is limited by the ability of the cylinder tube to resist buckling.

The main requirement is that the end covers cannot move when the cylinder is under internal pressure. With tie-rods, this can present problems on long cylinders, when one solution adopted is to add a support ring at the centre into which the tie-rods are studded. This effectively halves any deflection or elongation of the rods. Another solution is to use a central support ring which bows the full length tie-rods outwards slightly.

Pistons for light and medium duty cylinders are usually aluminium alloy castings. Aluminium castings or forgings are used for heavy duty cylinders with more robust sections. Alternatively, medium and heavy duty cylinder pistons may be fabricated from cast iron, brass or bronze.

The piston itself may be one piece, with machined grooves for ring seals, or two or three piece construction depending on the type of seals used (*eg* some types of seal require the piston to be disassembled for fitting). Cups, U-rings and C-rings are the favoured types of seal for medium and heavy duty pistons and for all larger diameter sizes. Wear rings may be incorporated in the piston assembly in addition to seals, and/or the piston length increased for better support in the case of long-stroke cylinders.

For satisfactory seal life a fine cylinder bore finish of the order of 15 microinches or better is required, particularly when using elastomeric seals. Finer finishing is desirable on cylinders intended for low operating speeds. Hard chrome plating followed by honing and polishing may be used to protect cylinder bores.

Piston rods are normally 050M40 steel (or a similar mild steel specification) ground and polished and chrome plated. Chrome plating provides suitable hardness, but hardened steel rods may be employed in some designs. Stainless steel, ground and polished, is now more favoured as an alternative. Regardless of the material used, a high finish is essential in order to minimize wear on the rod seals. This also favours the use of a non-corrosive rod surface.

Because of the relatively low thrust forces involved, rod diameter:cylinder diameter ratio is considerably lower than that of hydraulic cylinders; this has the advantage of reducing the piston area loss on a double-acting cylinder, as well as reducing rod weight (and thus the tendency to sag). However, the rod diameter size is governed by end load and rod length, the diameter being chosen so that the critical length of the rod is not exceeded.

For simplicity of production, rod diameters are usually standardized in smaller numbers than cylinder sizes — *ie* one 'standard' rod size may serve two or more 'standard' cylinder sizes. Alternative (larger) rod sizes may be needed for specific duties, and particularly for longer strokes. Loading limits for standard rods are also specific, although not always direclty given in the cylinder specification. All heavy duty cylinders require larger rod:piston diameter ratios than other types. Long stroke cylinders also present a particular problem since, as well as requiring a larger rod:piston diameter ratio to prevent buckling under load, the weight of a long rod will tend to make it sag in the case of horizontal cylinders. If this weight problem is likely to be critical, a hollow (tubular) rod can be used. It is usual in the case of longer stroke cylinders to find that the length of the rod bearing is increased to provide better support.

Rod bearings used range from simple plain bearings in a cast iron end cover to inserted metal sleeves in bronze or sintered bronze. The bearing assembly also invariably incorporates the rod seals. Simple seals are suitable for light duty cylinders. Medium and heavy duty cylinders normally employ multiple seals or seal sets. A separate wiper seal may also be incorporated at the outer end of the bearing assembly.

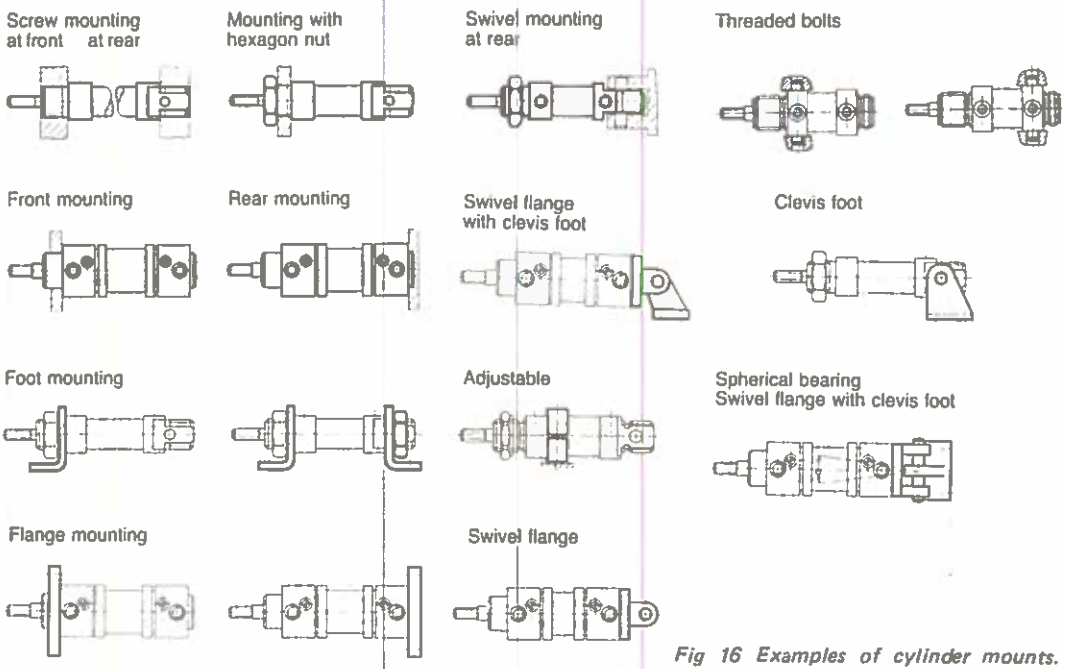


Fig 16 Examples of cylinder mounts. (Festo Pneumatic).

Air Cylinder Mounts

The method of mounting an air cylinder can affect the maximum rating or maximum thrust loading of the cylinder. Choice of mount may also be affected by the construction of the cylinder, *eg* the rod construction may favour tie-rod mounting as the most straightforward choice; or the construction may embody specific mounts — *eg* integral centreline mounts. In general, however, whatever the construction, standard ranges of cylinders are designed which take attachments to adapt any cylinder to a variety of different mounts (*eg* see Fig 16).

Mounts can be classified under three basic types:

(i) rigid mounts which absorb the force on the cylinder centreline.

(ii) rigid mounts which absorb the force on a plane parallel to, but removed from, the cylinder centreline.

(iii) pivot mounts.

Category (i) mounts are generally to be preferred where the forces to be absorbed are heavy. They also usually provide the most rigid form of mounting with a minimum of complication. Provided there is no possibility of rod misalignment, the mounting bolts are only stressed in simple tension, with no compound stresses.

Fixed category (i) mounts derived from cylinder end positions will normally exhibit a tolerance for some rod misalignment at the end of the cylinder stroke, but little or none at the other end of the stroke. With long stroke cylinders it may therefore be desirable, or necessary, to provide additional mounting support for the free end of the cylinder to prevent sagging and imparting compound stresses to the mounting bolts. Double mounts are always desirable where a long stroke is completely overhung and very long cylinders may benefit from an additional centre support.

Category (ii) mounts are often the most convenient, but have distinct limitations where heavy forces are involved, and particularly heavy shock loads. Such loads can tend to distort the cylinder, imparting heavy compound stresses to the mounting bolts. The platform to which the mounts are bolted may also need stiffening or support to resist bending loads which may be transmitted to it through the mounts, caused primarily by distortion of the cylinder.

A possible method of overcoming this is to fit keys or pins to take the shear loads, leaving the mounting bolts in simple tension. Sometimes a thrust key can be incorporated on the cylinder itself (see Fig 17), which can be fitted directly into a keyway cut in the surface of the mounting member. When using such methods of accommodating shear loads only one end of the cylinder should be keyed in order to allow for expansion and contraction due to changes in temperature and pressure.

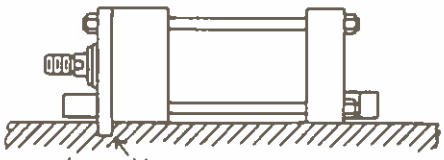


Fig 17

A category (iii) mount is normally required where the piston rod is required to traverse a curved path rather than a straight line. It may also be used to accommodate a degree of rod misalignment where the motion is basically linear.

Where the curved motion or misalignment motion is confined to one plane, simple pin mounts such as are provided by a trunnion or clevis are satisfactory, the pin being mounted on the centreline of the cylinder. A clevis or tongue and bracket mount is invariably fitted to the blind end of the cylinder. Trunnion mounts can be at either end or in the centre of the cylinder.

Centre or rod end trunnion mounting is generally preferred on long stroke cylinders as it has a greater resistance to rod buckling. Pins are normally designed to resist plain shear loads and thus trunnions or clevis ends should not be fitted with self-aligning bearings as these could introduce bending loads on the pins. Pivot mountings incorporating universal joints or swivel joints are produced, however, and may have to be used where the path described by the rod traverses more than one plane.

Rod ends can be plain or screwed, clevis or tongued, or fitted with a flange. Alternatively, a universal ball joint fitting can be used to accommodate rod movement in more than one plane. Similar considerations apply to those mentioned previously — *ie* self-aligning bearings should not be used to support pinned rod end joints where the pin is only designed to accommodate plain shear loads — see also Fig 18.

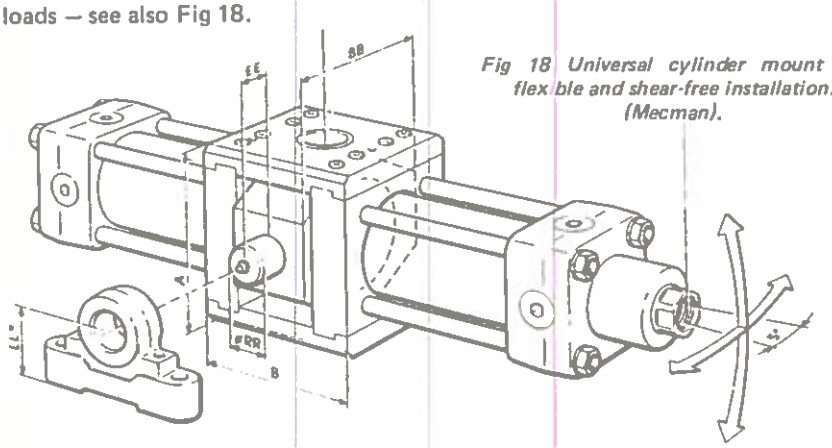


Fig 18 Universal cylinder mount for flexible and shear-free installation. (Mecman).

TABLE III — CYLINDER MOUNTS

Mount	Position	Remarks
Flange	(i) Rod end (ii) Blind end	Normally the most rigid and strongest method of mounting since mounting bolts are in simple tension, provided no misalignment of rod is present (or possible during operation).
Foot	(i) Single (ii) Double	In centre of cylinder — for light loads only. Rigid mounting but mounting bolts are under compound stress. A common and often the most convenient method of mounting a plain cylinder.
Pedestal or lug	(i) Single	Similar to flange mounting but heavy bending loads and compound stresses are transferred to base of pedestal and base fixing bolts.
	(ii) Double	Similar to double foot mounting.
Centreline	(i) Single (ii) Double	Suitable for light duties only. Rigid mounting which absorbs force on cylinder centreline. Mounting bolts are stressed largely in simple shear.
Tie-rod	(i) Rod end (ii) Blind end	Identical characteristics to flange mounting.
Trunnion	Single (centre or end)	Pivot mounting which absorbs force on cylinder centreline. Trunnion fixing bolts are in compound stress (tension and shear).
Screwed nose	Rod end	Rigid mounting with simple stressing.
Clevis	Blind end	Pivot mounting allowing for some misalignment of the rod whilst absorbing force on cylinder centreline.
Tongue and bracket	Blind end	Similar to clevis but with bracket fixing bolts in compound stress (tension and shear).

For cylinders of tie rod construction, mounting on the tie rods themselves provides a category (i) mount and is particularly attractive where mounting space is limited. An additional plate may, however, be added for flange mounting for ease of alignment. Further possibilities are summarized in Table III.

Energy Saving with Pneumatic Cylinders

The useful energy produced by an air cylinder is directly proportional to the product of the air pressure and swept volume. At the completion of the stroke the air, still under pressure, has 'expansion energy' which is normally lost energy since the air escapes directly to atmosphere. The amount of energy lost in this way depends on the initial air pressure, and in the case of 6—7 bar pressure can be as high as the useful energy produced in the cylinder, *ie* the *pneumatic* efficiency of a typical cylinder can be as low as 50%.

Whilst it is not possible to recover expansion energy as such, considerable savings in air consumption is possible using pneumatic energy-saving fittings mounted on the outlet ports of the control valve. In effect these regulate the air supplied to the cylinder to the minimum required for proper functioning, whilst still providing free passage for exhaust air. As well as savings in air consumption this can also improve the pneumatic efficiency of the cylinder since it may operate quite satisfactorily with a lower air pressure set on the energy-saving fitting. This is particularly advantageous where the air cylinder is over-sized for the specific duty requirement.

The principle of operation of the Legris EP 7000 energy-saving fittings is illustrated in Fig 19. On the outward stroke (Fig 19A) system pressure P1 passes through the control valve to the fitting. The closed position of the non-return skirt is maintained by the pressure, whilst at the same time the calibrated valve rises and lets through the amount of air necessary to drive the piston at a reduced pressure of P2.

At the end of the stroke (Fig 19B), pressure P2 together with the force of the calibrated spring closes the valve. The piston is maintained in position by pressure P2. For the return stroke (Fig 19C) the control valve is reversed, pressure P2 is exhausted causing the non-return skirt to open. Throughout the return stroke the air from the cylinder thus flows through this non-return valve.

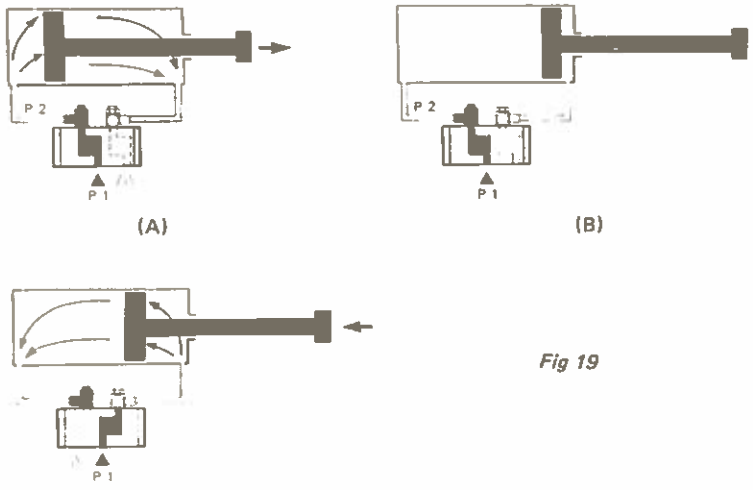


Fig 19

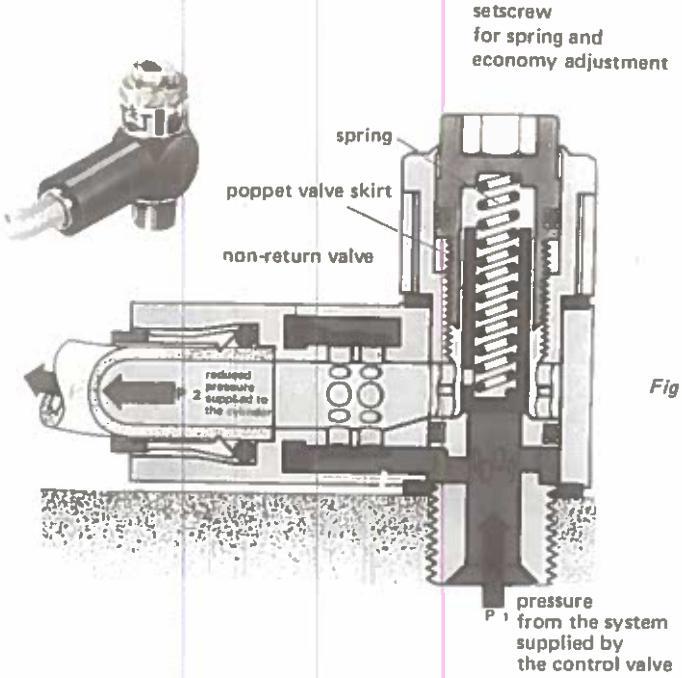


Fig 20

The construction of this particular fitting is shown in Fig 20. It includes a poppet valve the spring of which is calibrated by means of a set screw to regulate the air from a pressure of P1 to a lower pressure P2. This valve allows a passage of air at pressure P2 and closes when pressure P2 and the spring force balance out P1. The saving in energy is proportional to the difference between P1 and P2, set manually by adjustment of the set screw. Optimum setting is attained on a trial-and-error basis.

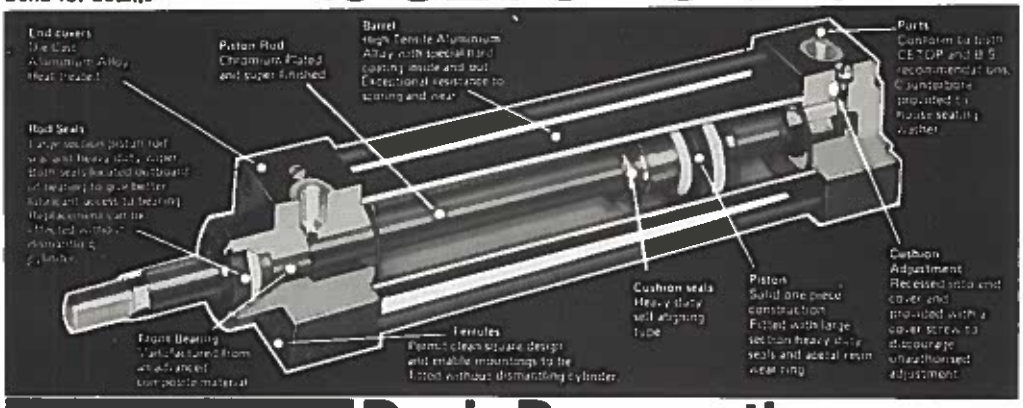
MODELS AVAILABLE:-
Aluminium (illustrated) —
Steel — Brass —
Stainless steel.

OPTIONS INCLUDE:-
Electrical reed
switch sensing,
viton seals hydraulic,
non lube and specials.

Send for details

Quick Delivery of Davis Air Cylinders to CETOP Standards

32mm
— 320mm bores.



DAVIS RED RING

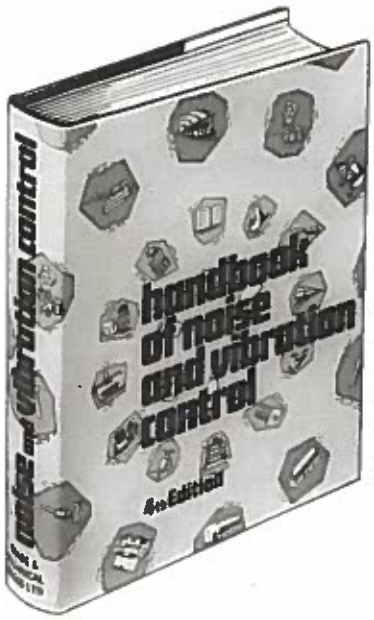
Davis Pneumatic Systems Limited

Plympton, Plymouth,
Devon, PL7 4BQ
Telephone: Plymouth
(0752) 336421
Telex 45657

order your copy now

- Completely revised and up-dated
- Handbook of vital importance to industry and commerce.
- Over 800 pages, 2,500 illustrations, diagrams, charts, tables, etc.

TRADE & TECHNICAL PRESS LIMITED,
CROWN HOUSE, MORDEN,
SURREY, SM4 5EW, ENGLAND.



The big M

Pneumatics–
Hydraulics–
Systems

The big M

We're available for any duty – from light to heavy.

MECMAN

MECMAN LTD.,

Mecman House, Suttons Park Avenue, Reading, Berkshire. RG6 1AZ
Tel: (0734) 664456 Telex: 848410

P. H. 02.5.2

Air-Hydraulic Cylinders

TWO-FLUID systems involving separate but inter-connected air and hydraulic (oil) cylinders fall into this category

(i) *Check units* — where a small hydraulic cylinder (or cylinders) is mechanically connected to an air cylinder for accuracy of control of cylinder movement.

(ii) *Air-hydraulic cylinders* — usually a back-to-back tandem arrangement of air and hydraulic cylinders with a common movement.

(iii) *Intensifiers* — where a low pressure air cylinder is used to generate a much higher pressure in a smaller hydraulic cylinder.

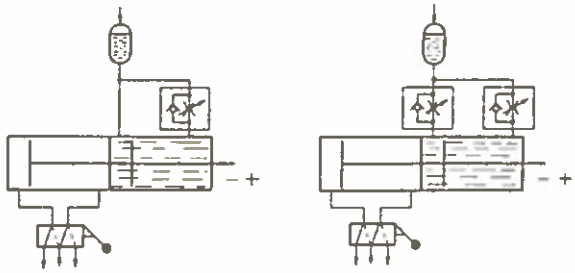


Fig 1

Hydro-Check Cylinders

Two examples of basic hydro-pneumatic cylinder circuitry are shown in Fig 1. In the left hand diagram speed is controlled in the + stroke of the cylinder. If the – stroke is to be controlled, the non-return restrictor valve is inserted in the other hydraulic cylinder line.

The second diagram shows the same circuit adapted to speed control in both directions. However, one throttle valve may substitute for both non-return restrictor valves where equal speed in either direction is called for, or can be accepted.

See also chapter on *Air Cylinders.*

Air-Hydraulic Cylinders

Typical applications of air-hydraulic cylinders include feeds for drills, milling machines, circular saws, *etc* and similar operations where the power supply available is compressed air, output forces required are relatively light, but a rigid feed is required.

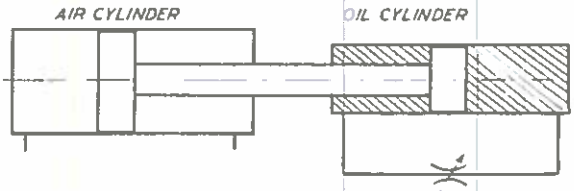


Fig 2

Hydro-pneumatic working is achieved by mechanically coupling a pneumatic cylinder and a low pressure hydraulic cylinder so that they have a common movement normally initiated by compressed air applied to the air cylinder, which is a double-acting type. The normal configuration is back-to-back mounting with a common piston rod — Fig 2. The hydraulic (oil) cylinder is also double-acting, but with a closed loop circuit. Control of the rate of flow of oil through this circuit governs the speed of operation of the tandem cylinder.

In practice, it is essential that air cannot leak into the hydraulic circuit, otherwise this would cause 'spongy' operation. Thus two sets of rod seals are employed, one on the air side and one on the oil side. Provision also has to be made on the hydraulic side to accommodate any changes in working volume of the oil — *eg* by the entry and withdrawal of the rod in the simple system of Fig 2; and also oil volume changes due to ambient temperature changes.

A logical design configuration is to make the air cylinder the blind end of the combination so that the hydraulic cylinder then has a through-rod giving equal displacements with either direction of motion. It is then possible to reduce the volume of the compensating oil reservoir to minimum proportions, or even eliminate it entirely if the cylinder will be working at a substantially constant temperature.

Speed Control

Further controls can be provided in the pneumatic circuit, which is the 'working' circuit. However, unloading or pressure relief valves cannot be used in the air circuit, since these could be operated by the damping produced by the hydraulic circuit, causing loss of working pressure. If unloading is required on the air side, this would have to be done via microswitches operated by mechanical movement, or suitable pilot valves.

Throttling may be provided on both sides of the hydraulic piston for adjustable speed control in both directions, or on one side only with a flap valve for differential speed control with fast movement on the opposite stroke, as in Fig 3.

Basic movement controls possible with the control unit(s) located in the hydraulic circuit are:-

Speed, one direction — one throttle valve or restrictor.

Speed, both directions — two throttle valves or restrictors.

Fast Approach, with final controlled approach —

(i) throttle and bypass valve in parallel

(ii) mechanical (applicable only in case of separate cylinders).

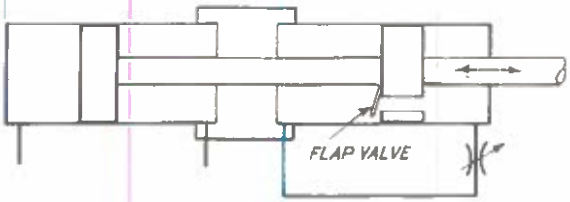


Fig 3

Fast Return — flap valve in piston.

Fast Approach, controlled return — flap valve in piston plus throttle valve in return flow line.

Controlled Approach, fast return — throttle valve in one line; flap valve in piston.

Various other combinations of the aforementioned are possible, *eg* fast approach with controlled final approach, followed by fast return. In that case the control units required would be those for both types of movements, on appropriate sides in the hydraulic circuit — see Fig 4.

In general:-

Throttle valve (or restrictor) provides speed control (in appropriate line).

Throttle valve with bypass valve provides differential speed control (in appropriate line).

Flap valve in piston provides uncontrolled fast approach, or fast return (depending on which side of the piston it is fitted).

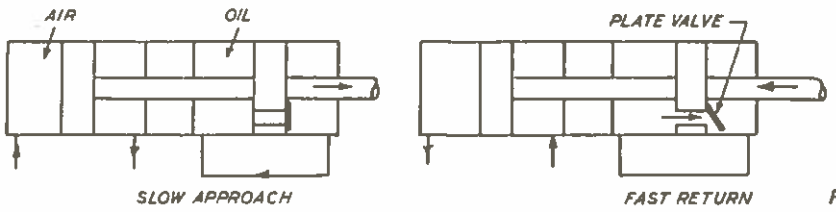


Fig 4

Intensifiers

Intensifiers are normally a special type of air-hydraulic cylinder whereby low pressure air can be used to generate a supply of high pressure fluid. The high pressure fluid may be air (or a gas), or a liquid (*eg* hydraulic oil or water). Intensifiers may be used directly as a source of low volume high pressure fluid, or as high pressure hold devices. In the latter case initial pressurization is achieved from a pump, the system then being switched over to the intensifier to produce and hold the final high pressure.

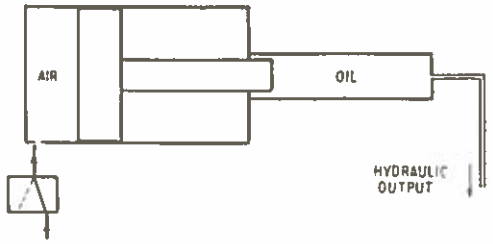


Fig 5

A diagrammatic representation of a typical air-hydraulic intensifier is given in Fig 5. Air pressure is used to drive the piston in a low pressure cylinder, the piston rod forming a plunger for a very much smaller diameter high pressure cylinder. Pressure intensification is directly related to the ratio of the cylinder bores, a typical intensifier ratio being of the order of 50:1. This would yield a hydraulic pressure of the order of 350 bar (5000 lb/in^2) from a conventional air line supply at 7 bar (100 lb/in^2). If necessary, very much higher pressures can be obtained from air-hydraulic intensifiers. Normally a manufacturer would have a basic air cylinder and vary the diameter of the hydraulic cylinder to give the desired high pressure output. Output volume, of course, reduces with reducing diameter of the high pressure cylinder for a given stroke (available

from the air cylinder), as well as the speed of reciprocation. Speed falls off as pressure builds up, until reciprocation finally ceases when the loads on the low pressure piston and high pressure plunger are balanced. Reciprocation of the air cylinder is achieved by any suitable form of valve gear.

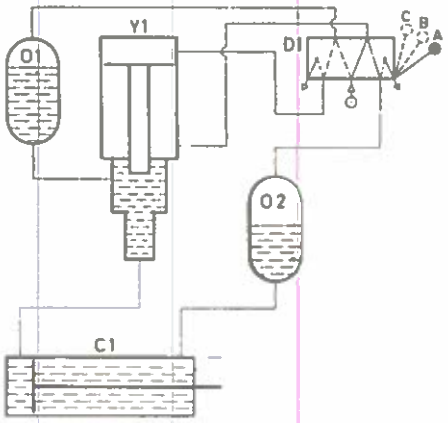
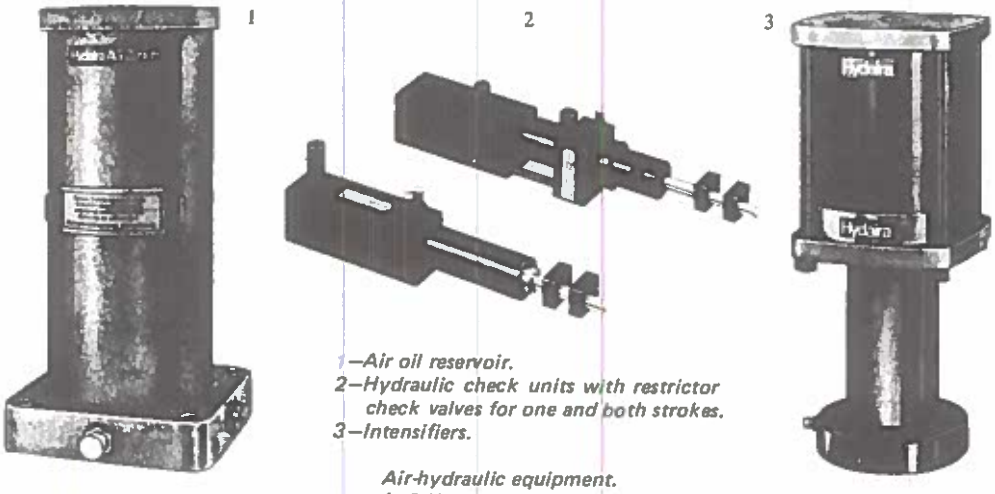


Fig 6

A typical circuit using a 6-way directional control valve is shown in Fig 6. In the starting position D1 is in the position A. C1 and Y1 are in the withdrawn position. When D1 is moved to position B, reservoir D1 is pressurized and simultaneously reservoir 02 is vented. C1 moves outwards under low pressure. When D1 is moved to position C, Y1 moves downwards under air pressure, and hydraulic pressure actuating the cylinder C1 is intensified. Return movement is accomplished by pressurizing 02 and venting 01.

See also chapter on *Air-Hydraulic Circuits.*



1—Air oil reservoir.
2—Hydraulic check units with restrictor check valves for one and both strokes.
3—Intensifiers.

Air-hydraulic equipment.
(1,3 Hydaira AG, Zurich;
2 Festo-Pneumatic).

Actuators

Bellows Actuators

A BELLOWS actuator works in the reverse mode to an air spring and has a similar geometry. Air springs connected to an external pressure source can, in fact, be used as actuators, the most suitable form being the convoluted bellows with one, two, or possibly more convolutions depending on the stroke required – Fig 1. Nominally, and dependent on the detail design, the maximum stroke obtainable from a single convolution bellows is of the order of ¼ of its overall diameter or less; of the order of ½ the diameter with two convolutions; and rather more than ½ the diameter with three convolutions.

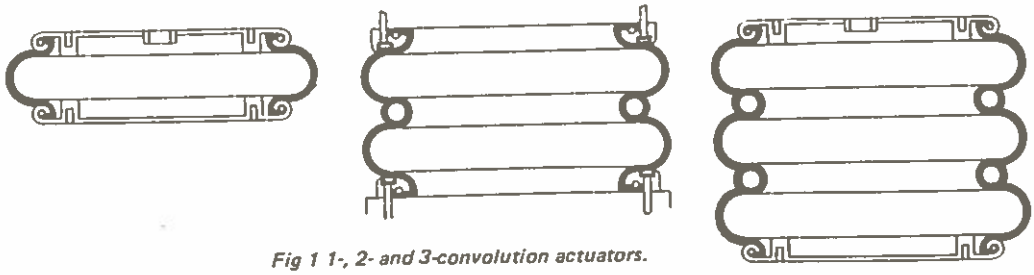


Fig 1 1-, 2- and 3-convolution actuators.

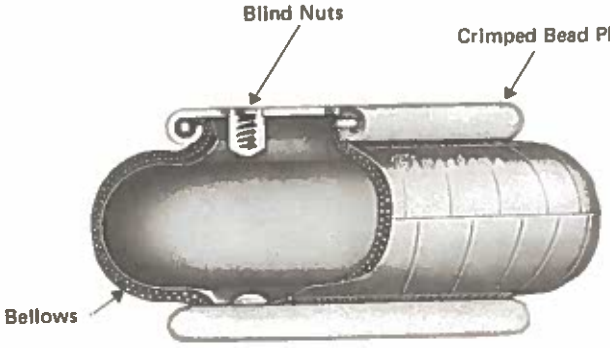


Fig 2 Firestone airstroke actuator

Typical bellows construction is shown in Fig 2. The bellows are formed from two plies of nylon reinforcing cord encased in synthetic rubber (neoprene), the cord being designed to provide an interlocking effect as internal pressure is applied. Crimped head plates provide hermetically sealed ends as well as a convenient mounting and accommodation for the air entrance port.

With this form of bellows the effective area of the actuator varies with extension, thus the force characteristics vary through the stroke range. Design can, in fact, be varied to provide different force-stroke characteristics, even constant force if required. A further feature of such actuator types is that they can accommodate considerable irregularity, thus dispensing with the need for a clevis in many applications where this working condition exists.

Rotary Actuators

The two basic forms of pneumatic rotary actuator are:-

(i) The torque cylinder or semi-motor.

(ii) Single or paired diaphragm cylinders.

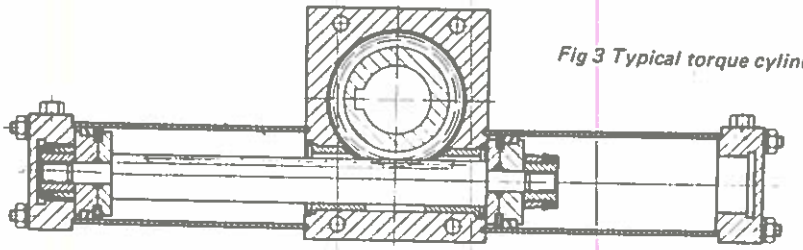


Fig 3 Typical torque cylinder in section.

The typical form of a torque cylinder is shown in Fig 3. It comprises two cylinders, back-to-back, with a common piston rod. The rod is cut with, or carries, a toothed rack engaging with a pinion mounted transversely in the centre section. Linear movement of the piston is thus translated into limited rotary output. Rotary movement is usually restricted to a maximum of 360°, or less. Nevertheless this is considerably more than that possible from a conventional cylinder connected to a lower output where the maximum angular displacement available is usually 120° or less.

A particular advantage with the torque cylinder configuration is that for constant applied pressure the torque output is constant over the whole range of rack travel. Cushioning may or may not be incorporated at each end of the cylinder, depending on requirements.

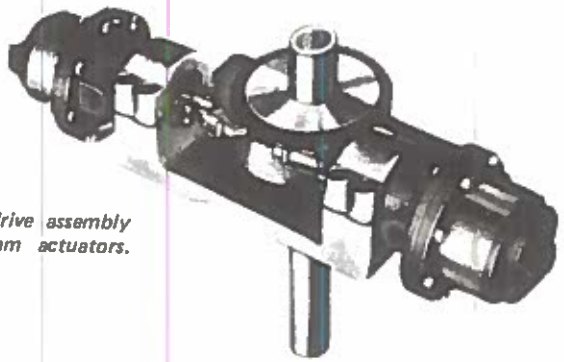


Fig 4 Telektron rotary drive assembly powered by two diaphram actuators.

A paired cylinder diaphragm-type rotary actuator is shown in Fig 4. Here two conventional diaphragm cylinders are mounted on a common centrepiece carrying a toothed wheel. Each cylinder rod terminates in a ratchet engaging this wheel. The selected diaphragm cylinder is operated in an automatic oscillating mode, driving the toothed wheel continuously in a series of progressive rotary movements. Selecting the opposite cylinder reverses the direction of motion.

Since diaphragm cylinders are essentially high-thrust devices, high output torques can be realized. Also this type of rotary actuator is capable of continuous rotation in either direction.

Rotating Cylinders

Two basic types of cylinder can be described under this heading. The true rotating cylinder is one where cylinder and piston are locked in such a manner that no relative rotary motion can occur between them. The complete cylinder is then mounted on a distribution gland which allows the cylinder to rotate relative to a workpiece. The other type of 'rotating cylinder' is one where the design provides for the piston to rotate freely relative to the cylinder with no loss of working characteristics. This implies that the piston seals must be capable of accommodating rotary as well as sliding motions.

Air Motors

AIR MOTORS are not direct replacements for electric motors or hydraulic motors. Rather they can be considered as an alternative choice in low power applications — *ie* under 15–22 kW (20–30 horsepower) where the specific properties and advantages of air motors can be properly utilized. Larger air motors are, however, coming more to the fore — see also Table I.

TABLE I — AIR MOTOR CHARACTERISTICS

Motor Type	No Load Speed rev/min	Typical Operating Speed Range rev/min†	Torque	Torque Curve	Initial Torque	Sizes (Typical Productions)
Vane	up to 20 000	7–15 000	Low	Flat1	V.low*	75 W up to about 15 kW
Piston	up to 2 000	600–1 500	Medium to high	Flat to steepish	High steepis	75 W up to about 20 kW
Diaphragm	up to 100	0–20	Extremely high	Steep	Extremely high	Under 100 W
Small turbine	up to 1 120 000	50 000–80 000	Extremely low	Flat	Negligible	75 W – 2 kW
Large turbine	up to 50 000	20 000–30 000	High	Flat	Higher than vane	3.75 – 70 kW
Gear	up to 3 000	1 500 – 2 000	Low	–	Low	0.75 – 15 kW
LSHT (lobe rotor)	Low	80–160	High	Peaking	High	–

†Direct drive or spindle speed

* Depends on number of vanes

Air motors can be operated continuously for long periods, provided they have an adequate air supply and satisfactory lubrication. They are very unlikely to overheat because the expansion of exhaust air has a cooling effect. They can operate at the extremes of ambient temperature, although for sub-zero temperature working special seals may be required. They can be operated

under water, down to depths of 12 m (40 ft) or more; in this case vents and exhaust must be led to the surface to avoid excessive back pressure being developed, and additional seals may be necessary to prevent water entering the motor.

Power is normally compressed air from a compressor or compressed air line. For short-term running the motor may be fitted with its own air supply in the form of an air receiver — this is only viable if high pressure air is available as otherwise the size of receiver required would be too large.

Air motors can also be operated by any other clear gas at a suitable pressure, provided lubrication requirements can be met — *eg* producer gas or even process gases where a suitable differential pressure can be extracted from the system without interfering with the process requirements.

Types of Air Motors

The most widely used types of air motors are the rotary vane and radial/axial piston motors. Other types include gear motors, turbine motors and diaphragm motors.

In addition to general purpose motors (vane and piston types particularly), air motors are commonly produced for specialized duties — *eg* tool drives, valve actuators, starters for internal combustion engines and gas turbines, winches, hoists. In addition, the field of application of air motors is partially overlapped by rotary and semi-rotary actuators, and air motors may also be used as drivers for rotary, semi-rotary and linear actuators.

Vane Motors

Rotary vane motors are comparatively simple in design and construction and widely used in a range of sizes from 75 W (1/10 hp). They are extremely compact, with a high power to weight ratio and free running speeds up to 15 000 rev/min or more.

1 — Cylinder
2 — End plate
3 — Ball bearing
4 — Silencer
5 — Planetary gear
6 — Motor housing
7 — Rotor
8 — Vane
9 — Air inlet to cylinder

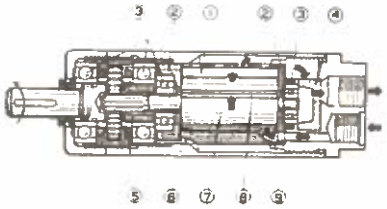


Fig 1 Vane-type air motor.

A section of a typical vane motor is shown in Fig 1. Four vanes are usual, although some motors may have more. The greater the number of vanes, the less the internal leakage and the higher the starting torque, although friction and wear are increased. Good starting torque also depends on the vanes sealing at low rotational speeds. This is normally provided by 'spring loading' the vanes in a radial direction, either pneumatically or mechanically.

Torque generated by a vane motor is equal to the product of the applied air pressure, the exposed vane area between the inner motor and casing bore and the moment arm of this area. Both latter quantities are relatively small, so the vane motor is essentially a low torque motor, dependent on high operating speeds to generate good power. The supply pressure is a significant factor in determining the torque and power realized — see Fig 2.

If necessary, torque output can be increased by reduction gearing — see Fig 3. With 20:1 reduction gearing the vane motor will show substantially the same peak speed and peak speed

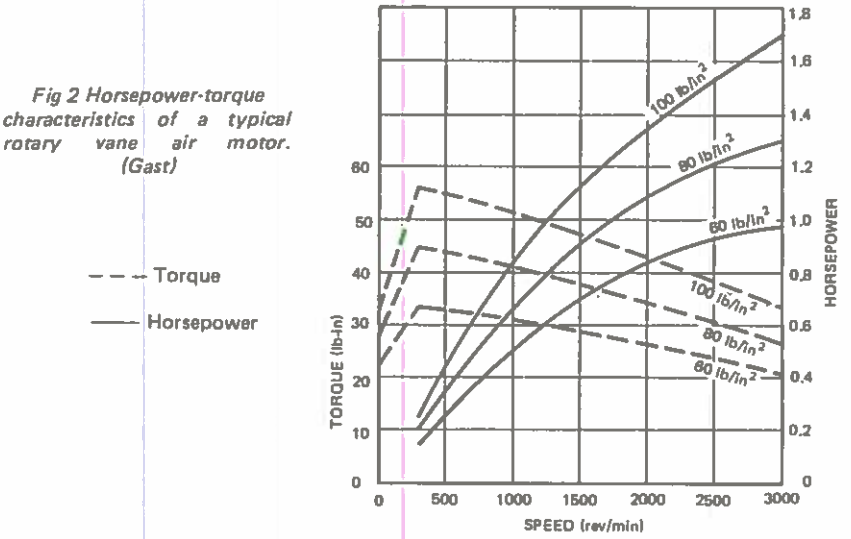


Fig 2 Horsepower-torque characteristics of a typical rotary vane air motor. (Gast)

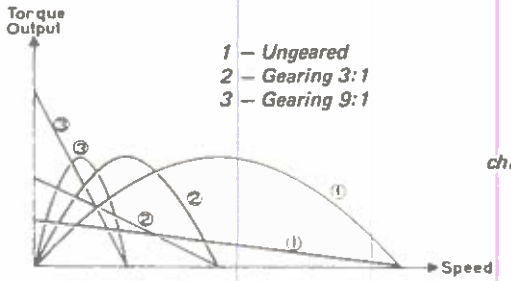


Fig 3 Diagram showing how the characteristics of an ungeared motor are changed with gearing. (Atlas Copco).

torque as a piston motor with higher efficiency (*ie* lower air consumption per horsepower). This, however, can be partially if not wholly offset by gearing losses. The main point is that the use of reduction gearing to adjust both speed and torque levels to specific operating requirements is very attractive with rotary vane air motors.

As shown in Fig 2, characteristically torque output is substantially linear, although the starting torque available is invariably lower than the theoretical maximum or stall torque. Also torque values are relatively low, good power output depending on high running speed. The peak of the bhp curve may occur at 50% of the no-load (maximum) speed or somewhat higher, depending on the detail design and, particularly, the porting. Motors of this type, in fractional horsepower sizes, commonly have free running speeds of the order of 20 000 rev/min, although some may incorporate governors to limit the no-load speed attained. Power-to-weight ratio is good although starting torque is low and low speed performance relatively poor. A vane motor would not, therefore, be used for applications requiring low and creep speeds.

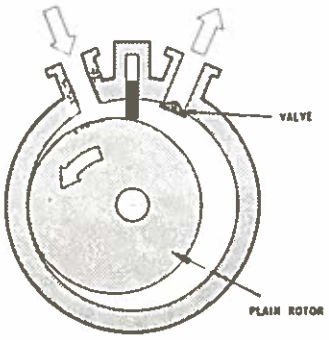
Chief applications of vane motors are:

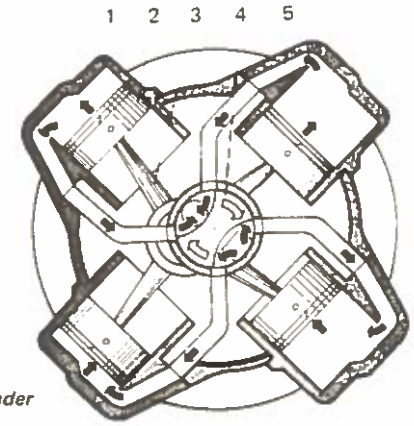
Fractional horsepower sizes — portable air tools, multiple light duty tools, *etc*.

Larger sizes — larger air tools, pump drivers and general driver duties where light starting loads only are involved.

Included under the general classification 'vane motors' are external vane motors, although these are less common. The cam-vane motor, shown diagrammatically in Fig 4, is based on a single vane rubbing on a solid rotor, thus avoiding the uneven wear of vanes running over port openings. It is far more limited in performance, however, with an inherent 'dead-spot'. Thus to provide continuous rotation, a flywheel type load is required; two or more such motors may be connected in series to promote smoother and continuous running.

Other variations include the use of lobe-rotor designs, literally in the form of pumps or compressors working in reverse. These have been specifically developed for high temperature work, but have little general application.





1 – Cylinder
2 – Piston
3 – Connecting rod
4 – Rotary valve
5 – Air channel to cylinder

Fig 5 Typical construction of a piston-type air motor. (Atlas Copco).

Piston Motors

Piston motors operate at much lower speeds than vane motors because of the greater inertia of the reciprocating parts. Free running speed is usually 3 000 rev/min or less, with maximum horsepower developed at 1 000–2 000 rev/min.

The majority of piston air motors are multi-cylinder units which may be in-line, vee, H, flat-four, radial or other variants. There are also further variations on conventional reciprocating engine design, such as crankless motors employing a rotating piston-cylinder assembly with slotted piston ends traversing an elliptic cam ring. In the main, however, the vee is the most popular configuration for four-cylinder motors, with the radial configuration equally popular for three or more cylinders. Radial configuration offers some possibility of weight reduction and is frequently adopted for larger motors. Either an odd or even number of cylinders can be used, *eg* 3, 4, 5 and 6 cylinder radials are all common types.

The vee-four motor normally has the two cylinder banks at 90° to each other. Drop forged two-throw crankshafts are normally used with two connecting rods on each crank pin. Distribution valves may be oscillatory or rotary, there being little to choose between them. Valve timing can be arranged to provide equal power in each direction of rotation, or may be biased to provide asymmetrical timing and increased speed and power in one direction of rotation.

The typical radial motor has all connecting rods mounted on a common crank – *eg* see Fig 5. Power and exhaust strokes are controlled by suitable distribution valves and a two-stage exhaust

is often adopted. In this case the inlet valve opens just after top dead centre and remains open for approximately 160°. The inlet valve then closes and a large primary exhaust valve opens quickly to release the high pressure air to the atmosphere. After a further 40° or so of rotation, this valve closes and a smaller exhaust valve opens to continue evacuation of the compressed air. Reversal of direction of rotation is obtained by interchanging the inlet and secondary exhaust connections. It should be noted that the radial configuration particularly lends itself to porting (including a shaft port) instead of the use of mechanically operated valves.

Axial piston motors normally have 4, 5 or 6 pistons, but are normally only made in smaller sizes – *eg* less than 2.5 kW (3½ horsepower). They are fast starting with high starting torque and reach full operating speeds almost instantly. See also Fig 6.

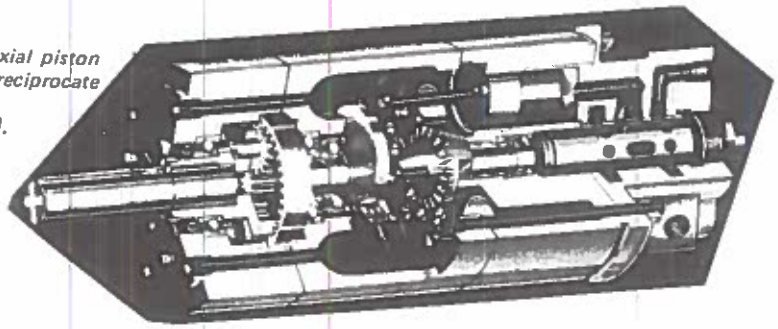


Fig 6 Cutaway view of an axial piston motor. The small pistons reciprocate axially in sequence. (Gardner-Denver Co).

All piston motors characteristically develop high starting and initial torque, falling with increasing speed. Both torque and bhp are directly dependent on supply air pressure. Starting, optimum running and stall torques are generally related to supply pressure as shown in Fig 7. Torque characteristics can be modified to some extent by detail design, *eg* valve timing and valve sizing. As

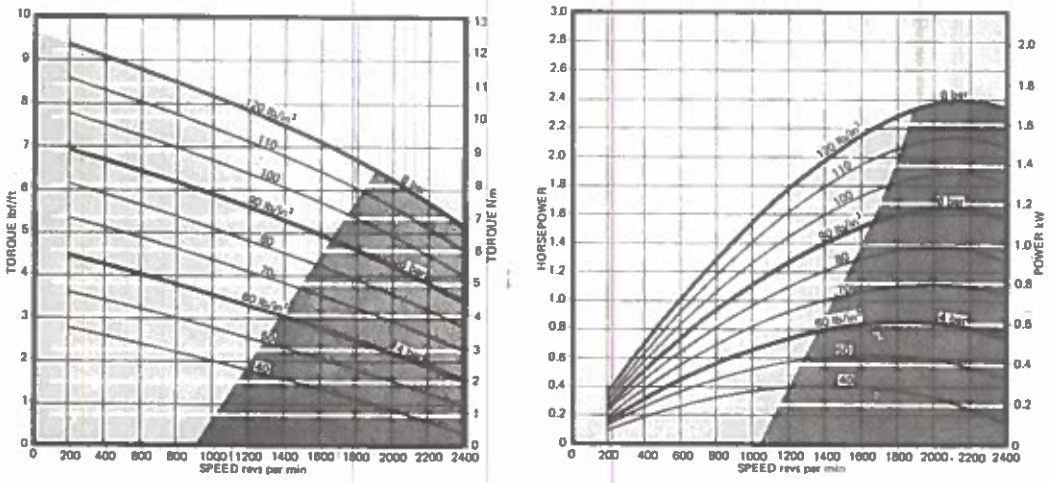


Fig 7 Characteristics of globe RM10 heavy duty radial piston motor. (Denver Fluid Power).

they are positive displacement machines, air consumption is directly related to speed, *ie* the air consumption/speed curve is substantially linear for any given supply pressure. Power output is of the order of 5—10 kW/dm^3 cylinder volume, depending on the size and design.

Running speeds are generally low to moderate, although smaller piston motors may be designed to run at high speeds. In this case speed reduction with torque multiplication may be incorporated via reduction gearing to an output shaft.

The overall characteristics of piston motors make them suitable for an extremely wide range of applications. With modification, standard designs can often work satisfactorily down to supply pressures as low as 0.35 bar (5 lb/in^2). Continuous lubrication is essential and the method of lubrication will depend largely on the individual design and the need (if any) for separation of the various lubricants used. Thus a lubricant containing an emulsifying agent may be desirable in cylinders to prevent 'water washing', but would not be suitable for splash lubrication in a crankcase.

Variations on piston motor designs include swashplate motors, face-cam motors and other forms of axial piston motor. In general such designs have been largely superseded by vane motors.

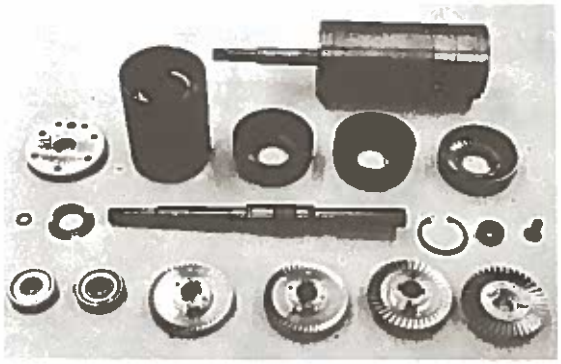
Turbine Motors

The small turbine is essentially a very high speed motor, capable of generating only minimal torque. Torque, in fact, is virtually negligible, except at mid-speeds, with maximum speeds of the order of 50 000 to 80 000 rev/min. Applications are thus distinctly limited. Torque multiplication by reduction gearing is not a viable proposition, because the load involved would be of the same order as the torque available. Hence the main application of small turbine motors is where very high speeds can be used with direct drive against comparatively light loads — *eg* high speed pencil-type grinders and dental burrs. Like other types of air motor a turbine can, of course, be stalled without damage. Some turbine motors may be fitted with mechanical governing devices to limit the maximum speed which can be attained.

Turbine motors operate with clearances throughout, with no sliding or rubbing parts other than the spindle bearings. Bearing friction is thus the limiting factor governing free running speed. Turbines can operate equally well on dry air (*ie* without lubricated air), provided the bearings are self-lubricating or have separate lubrication. This can be an advantage in specialized applications.

Impulse Turbine Motors

Originally developed in the 50 W — 2.25 kW (1/16 — 3 hp) range, very much larger turbine motors are now finding increasing application in sizes from 3.75—70 kW (5—95 hp). These are



137 shaft horsepower impulse turbine air motor
(Tech Development Inc).

normally single-stage impulse turbines coupled to a planetary gear reduction set. They are directly competitive with vane and piston motors, being smaller and lighter with the same size. They are normally used with reduction drive output, giving high stall and starting torques.

Diaphragm Motors

A diaphragm motor is basically a reciprocating device with limited stroke. Rotary output motion is derived via a ratchet and pawl — Fig 8. Essentially speed of working is low, but very high torques may be developed. For example, for the same amount of air consumption a diaphragm motor is capable of developing up to 800 times the torque of an ungeared vane motor, or up to 40 times the torque of a piston motor. Drive output is normally fractional horsepower, however, because of the low speed of working.

This diagram shows a typical valve chest associated with the diaphragm movement. The centre of the diaphragm is rigidly attached to a rod, one end of which forms the thrust shaft while the other admits high pressure air to the diaphragm chamber and then opens the chamber to exhaust as soon as the diaphragm has been displaced over its maximum travel. The return motion of the diaphragm may be driven by compressed air in both directions, *ie* high pressure air being admitted alternately to each side of the diaphragm, with corresponding provision to exhaust each side of the chamber in opposite phase. The power chamber is completely sealed, enabling the motor to be operated on very restricted air flows so as to produce extremely low speeds with no loss of thrust or torque. Further, since the thrust is always acting tangentially to the ratchet wheel, there is a little variation in the torque generated throughout a cycle, which enables the motor to be accurately positioned at any point on the working stroke.

Typical performance characteristics are as shown in Fig 9. Very high initial torque is available, torque falling off rapidly with increasing speed, although the speed range is strictly limited. Peak

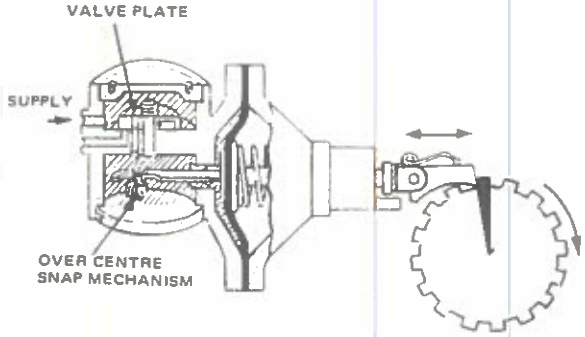


Fig 8

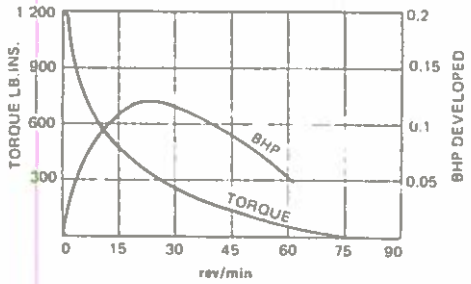


Fig 9

bhp is normally developed at circa 20 rev/min for motors of this type. In addition to the ratchet wheel diameter size variation (which affects speed as well as torque), motors of this type are also produced with two and three diaphragms operating on the same shaft to increase the thrust and final torque.

Diaphragm motors are particuarly suitable for high torque, low speed applications – *eg* for valve actuators. Here, high initial torque is required to crack open the valve. The falling load will then enable the motor to build up to a good running speed when the pressure differential across the valve has dropped. Additionally, if used as a valve positioned on an automatic control system, the maintenance of output with restricted supply air flow maintains full power operation of the valve as the balanced condition is approached. Diaphragm motors can, of course, also be used directly as vibrators.

Diaphragm motors are particularly efficient (although less so if the diaphragm is driven in both directions by pressurized air), and because of the null-leakage characteristics, air consumption is directly related to bhp or speed, but not torque.

Gear Motors

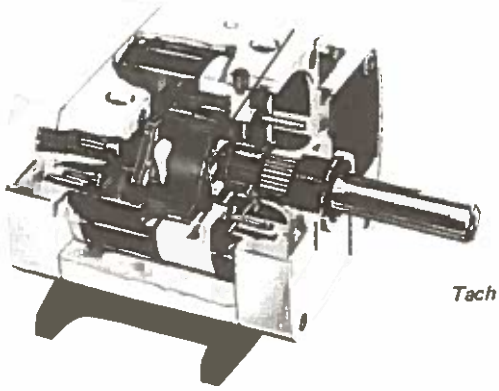
Meshing gear motors are a further type, but are not directly competitive with piston or vane motors. The relatively low efficiency (high leakage loss) of straight spur gears can be largely overcome by employing helical gears.

Gear-type air motors are generally characterized by moderate running speeds, typical operating speeds being of the order of 1 500 to 3 000 rev/min with direct drive. Output speed can be reduced and torque multiplication obtained by the use of reduction gearing. The application of gear-type motors is, however, strictly limited and few commercial versions are produced. An exception is the modern LSHT (Low Speed High Torque) motors developed specifically for pneumatics.

The description *gear motor* is also given to rotary vane motors fitted with multiple reduction gearing.

LSHT Air Motors

Low Speed High Torque (LSHT) air motors are a comparatively new development, derived basically from LSHT hydraulic motors of internal gear type with lobe rotors of special form (*eg* 'Gerotor'). The pneumatic equivalents are considerably modified in detail design and generally use different constructional materials (*eg* hard coated aluminium in conjunction with engineering plastics).



Tachyon® Gerotor cutaway illustration.

Primary characteristics are very low running speeds (20—160 rev/min) and positive displacement characteristics with very high torque (including high starting torque). Essentially, in fact, they can be considered as torque-generating motors rather than power drivers in the conventional sense.

Compressed Air Supply

Most air motors are designed to operate off standard industrial air supplies, with power ratings given for a supply pressure of 6.2 bar (90 lb/in^2). It is a basic requirement that the air supply be clean and dry to prevent possible motor damage. Commonly an air motor is used with a combined air filter, pressure regulator and lubricator immediately preceeding it in the branch air line to regulate and condition the air delivered to the motor.

Control of Air Motors

The control of any air motor is usually simple and direct. For most applications an ordinary throttling valve is adequate for variable speed control in conjunction with a four-way valve if reversing is required. This will provide full control of both speed and direction. Special control valves are normally employed, however, rather than standard air line control valves, in order to realize the optimum characteristics of the motor (*ie* to achieve progressive inlet and exhaust opening). Idling speed can be controlled by throttling or by a centrifugal governor which keeps the speed constant — see Fig 10.

Whilst the speed/torque characteristics are largely determined by the type of motor, most manufacturers offer a range of gearboxes for direct (or integral) coupling to their motors, thus

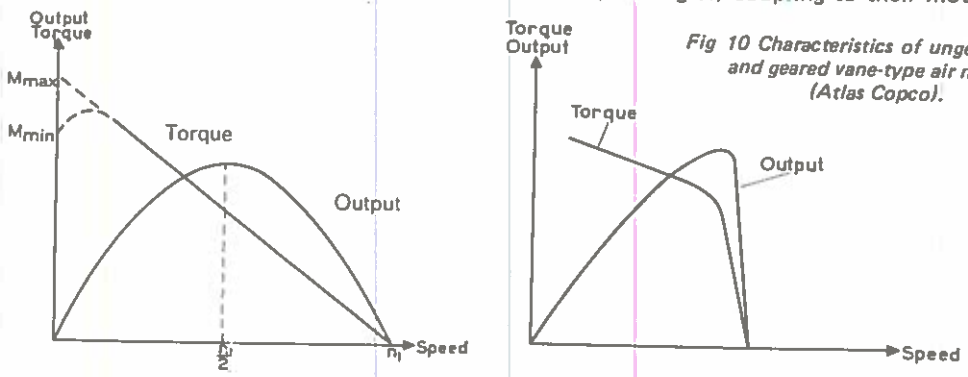
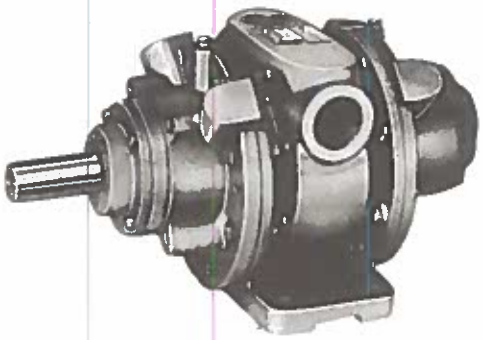


Fig 10 Characteristics of ungeared (left) and geared vane-type air motor. (Atlas Copco).



Gast model 16AM-FRV-2 face and foot mounting reversible air motor.

providing a range of speed/torque characteristics from a single unit. Another accessory commonly offered is an automatic brake for use in lifting, hoisting or similar duties.

There is no limit to the number of times an air motor can be started, stopped or reversed except that incurred by the speed of operation of the control valve. Air motors accelerate rapidly and will run as fast as the air supply and load resistance permit. Speed can be controlled by simple throttle or pressure regulating valves which adjust airflow rate or pressure respectively. The possible speed regulation is normally down to 10% or better of the full-throttle full-pressure speed against a given load. When controlled to any particular speed, however, actual running speed will vary if the load varies.

Exhaust

For normal industrial applications exhaust is vented directly to atmosphere via a silencer. In 'clean' areas it may be desirable or necessary to remove fine oil mist from the exhaust air by fitting a suitable filter. In ultra-clean areas the exhaust can be led outside the building. In this case a suitable size of exhaust line must be fitted in order to avoid excessive backpressure which would otherwise reduce the power of the motor.

The exhaust from an air motor is inherently noisy, which may call for the use of an exhaust silencer in particular applications. Absorptive type silencers offer the least back pressure (and thus lowest power loss). Other types used include conventional automotive-type silencers, dissipative silencers and phase-shift silencers. See also chapter on *Noise Control* .

Summary of Performance Parameters

Broadly speaking, the torque and power characteristics of an air motor can be regarded as similar to those of a permanent magnet or series wound d.c. electric motor, with maximum torque at stall, decreasing substantially linearly with speed. Maximum power is then achieved at around 75% of the free running (no load) speed. With the application of load, the running speed decreases (from

TABLE II – PERFORMANCE PARAMETERS

Motor Type	Maximum Power Developed at (%) No Load rev/min*	Speed at Maximum Power	Reduction Gearing Practical	Torque/ Speed Ratio	Speed Control	Torque with Falling Speed
Vane	45–55	High	Yes	Low	Air pressure	Rising
Piston	55–65	Low to moderate	Yes	Medium to high	Air pressure	Rising
Diaphragm	20–25	Very low	–	Extremely high	Air flow rate	Rising steeply
Turbine	75–90	Extremely high	No	Extremely low	Air pressure	Falling
Gear	50–60	Moderate	–	Low	Air flow rate and air pressure	Straight
Lobe rotor	60–70	Low	Essential	High	Air pressure	Falling
Linear		High†	–	–	Exhaust restriction	

*Approximate guide only †Without exhaust restriction

the free running condition) until the torque balances the load. With a constant load, speed can be further adjusted by variation in air pressure supplied. On the other hand, with varying load the motor speed will continually adjust to the load. These basic characteristics are modified in detail by the type of air motor concerned and the specific design characteristics – see Tables I, II and III.

TABLE III – AIR MOTOR – AIR CONSUMPTION FIGURES AT MAXIMUM POWER

Motor Type	Air Consumption per kW l/sec	Air Consumption per kW ft^3/min	Air Consumption with Increasing rev/min
Vane – small	30	63	Rising
Vane – large	17	36	Rising
Piston	18	38	Rising
Link-type piston	21–22	46	Rising
Turbine – large (geared)	8,0	1700	Constant

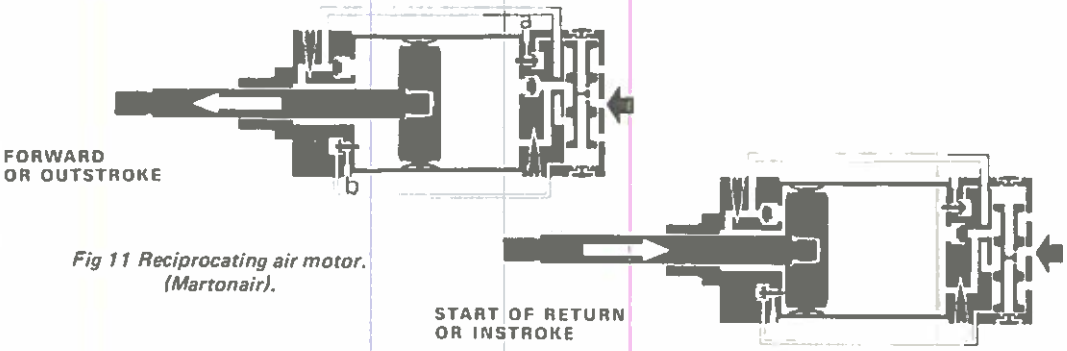


Fig 11 Reciprocating air motor. (Martonair).

Linear Motors

The linear motor is essentially a double-acting cylinder associated with suitable control gear to provide automatic reversal at the end of each stroke and thus continuous reciprocal motion. This control may be initiated by mechanical triggering, or pneumatic or electronic signalling (*eg* controlling a pilot-operated or solenoid-operated valve respectively) – Fig 11.

The reciprocating output of the cylinder (*ie* rod movement) may be used directly or to drive a crank for final rotary output. In the latter case, the crank throw must obviously be matched to the stroke available, or rather the crank throw will, in fact, govern the actual cylinder stroke achieved. The versatility of the linear motor may be further improved by providing variable speed of stroking. This is usually done by fitting adjustable restrictors on the exhaust side controlling the rate of escape of pressurized air from the non-pressurized side of the piston, and thus the back-pressure, which effectively controls the speed.

Strictly speaking, a linear motor adapted to rotary final output is no longer a linear motor as such. Connected to a crank mechanism it is really a single cylinder piston motor and its characteristics will be similar. If a short stroke cylinder is used, coupled to a ratchet drive, then its performance characteristics will approximate to those of a diaphragm motor.

See also chapters on *Lifts, Hoists and Winches* and *Turboexpanders.*

SECTION 2e

Ancillaries

Seals

SEALS FOR air cylinders and valves are not normally called upon to seal pressures higher than about 10 bar (150 lb/in^{1}). However if the fluid to be sealed is a gas, rubbing speeds tend to be high, and the seal may have to be operated under 'dry' conditions with minimal lubrication. Further adverse operating factors are the relatively low differential pressure available to 'energize' flexible lip seals, the deteriorating effect of oxygen on seal materials in general, and the high operating temperature that may be involved in certain pneumatic services.

Basically the requirement is for a high conformity, low friction seal for pistons and piston rods. This favours the use of O-rings for smaller cylinder sizes (and for valve seals), and the 'softer' types of pressure-energized seals for larger sizes. The air permeability of the seal material is also important.

In the case of a seal operating dry, or substantially dry, the apparent permanent set of an elastomeric seal may appear much higher (typically twice as high) than that of the same seal used as an oil seal. This may call for greater initial interference when assembling the seal, and subsequently higher initial friction during the first period of running when the permanent set is taken up. Selection of a suitable interference is more difficult, and errors may cause the seal either to continue to bind and develop high friction and wear or to run in a loose fit with consequent leakage. A permanently binding seal is more likely than a 'loose' seal, particularly if the seal is proportioned as a 'dry' type and subsequently lubricated by oil-mist or oil-fog. Only a proportion of the seal will operate under oil-mist or oil-fog conditions, unless provision is made for the amount of swell the seal is likely to undergo during its 'conditioning' period. This period may, in some cases, extend to several weeks of operation.

A period of immersion in fluid may also leach out a proportion of the plasticizer in the seal. If the seal is subsequently allowed to dry out it will shrink. For a dry seal, therefore, the material may have to be chosen specifically for a small amount of permanent set so that interference is not reduced too drastically in service. Seal sets and packings which can be adjusted can thus offer very real advantages in certain applications, although adjustment of packing pressure is not necessarily a satisfactory answer where consistency of performance is required. It may provide a means of adjusting sealing, but the friction will be modified both by the varying seal pressure and a variable coefficient of friction.

Specialized pneumatic seals are usually designed for simple assembly. In general the aim is to provide increased flexibility in the section design, with particular attention to conformation at the

seal lip to provide maximum retention of the small amount of lubricant generally found in pneumatic applications. Increased flexibility of the lip can also reduce friction. At the same time the section must be stable enough to prevent spiralling or 'roll over' when sealing reciprocating motions.

Further improvements can be realized by development of specialized elastomers, 'tailored' for optimum hardness, flexibility and resistance to compression set, and for maximum compatibility with the temperature, pressure and lubricants normally involved in pneumatic systems.

O-Rings

As a general rule O-rings are best suited for smaller sizes and lighter duties as reciprocating seals, although this is by no means an overall recommendation. O-rings have been used successfully both as rod seals and piston seals on relatively large cylinder sizes. The main requirement is that the O-ring be used within its rating or capabilities, and mounted in a properly matched groove.

The conditions most favourable to O-ring reciprocating seals are a short stroke; adequate lubrication; and moderate rubbing speeds. Conditions unfavourable to O-ring reciprocating seals are side and bending loads causing eccentricity of the moving system and high unit loads; very slow reciprocating speeds; and very low pressures accompanied by slight leakage. Both the two latter conditions tend to develop very high friction which can be damaging to the seal.

One basic limitation of the O-ring seal is that it normally performs best under conditions of adequate lubrication. With lack of lubrication rubbing friction can approach the static friction holding the ring in its groove. This can mean that some parts of the ring slide whilst others roll simultaneously, leading to excessive twist developing in the ring, promoting spiral failure.

U-Rings

The U-ring suitably proportioned and in a relatively soft rubber, is capable of providing particularly good sealing with low friction at low pressure. Desirable features are thin lips for response to pressure energization, generous length for good flexibility, with the lips themselves designed to have low lip loads for low friction and at the same time enough contact area and preload for good sealing. A square heel section is also preferable to minimize heel wear and 'nibbling' (Fig 1).

U-ring seals and various modified forms developed specifically for pneumatic services — *eg* rod and piston seals — therefore tend to differ in cross section from similar seals intended for hydraulic duties.

A modified form of U-ring developed specifically as a pneumatic seal is the Gasco H-7, shown in Fig 2. This has long, thin lips (for good flexibility), rounded lips for maximum retention of lubricant, and generous interference for good sealing.

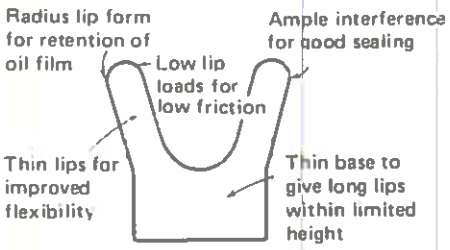

Radius lip form for retention of oil film
Ample interference for good sealing
Low lip loads for low friction
Thin lips for improved flexibility
Thin base to give long lips within limited height

Fig 1 Basic requirements for a U-ring pneumatic seal.

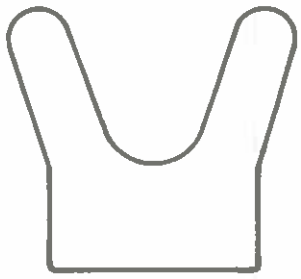


Fig 2

A considerably more elaborate section, which in effect is a rocking U-ring, is the Trist 'Pneuflex' seal, shown in Fig 3. This incorporates a Y-shaped section which can 'rock' about the base of the main U-section when assembled.

Trist 'Pneuflex' pneumatic seal.

Fig 3 'Pneuflex' pneumatic cylinder seal (Trist).

C-Rings

C-rings employ similar sections to U-rings, except that one leg is shortened and the other extended into a more flexible form of lip seal. This leg only provides the sealing action, hence C-rings are made in either external form (for rod seals) or internal form (for piston seals).

A number of proprietary rings have been developed from this section form, including more rigid sections for heavy duty working where the lip is formed by a V-shaped notch in the section and on the shorter limb. These are sometimes called automatic sections or distributor seals. They can be used singly or in sets. In the latter case their rigidity may be insufficient for them to be fitted without headers. True automatic sections (distributor seals) work on the principle of a compression seal, the sealing face being the solid limb opposite to the lip, although the section is also pressure energized — *eg* see Fig 4.

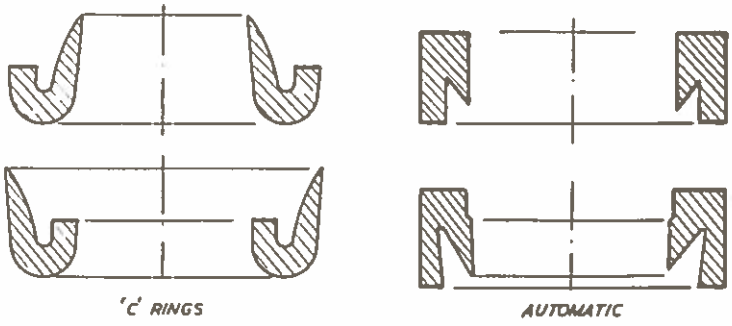


Fig 4

Special Sections

Modified U-ring forms, C-rings and modified C-rings, and other special sections may also be used for piston and rod seals on pneumatic cylinders. A number of these have been specifically developed for pneumatic services. An example described as a *caliper* seal is shown in Fig 5. This is intended as a direct alternative to an O-ring, fitting in a simjlar groove, and having similar dynamic

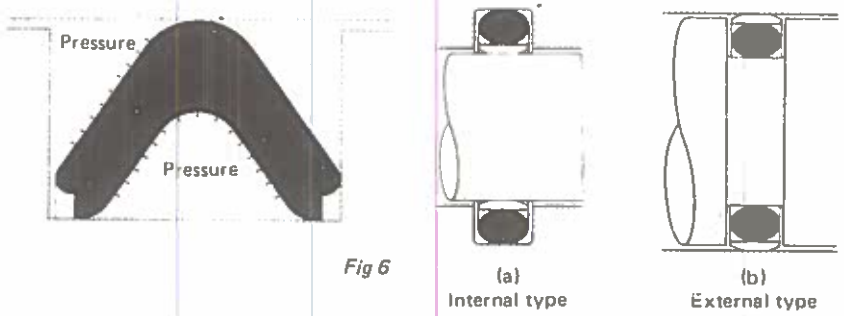


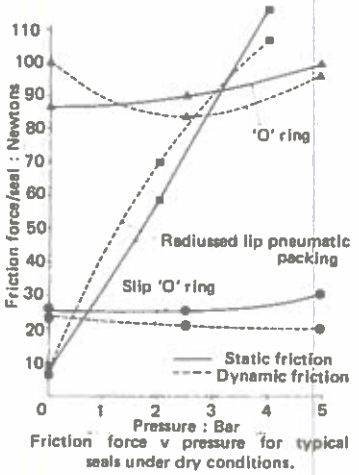
Fig 6

friction to an O-ring, but much lower break-out friction because of the much lower cross section nip and greater flexibility. Small grooves on the inner diameters of the seal allow air to enter the seal and actuate the opposite lip against the side of the groove. The seal is thus fully pressure-energized under both static and dynamic conditions, and will seal in either direction.

Composite Seals

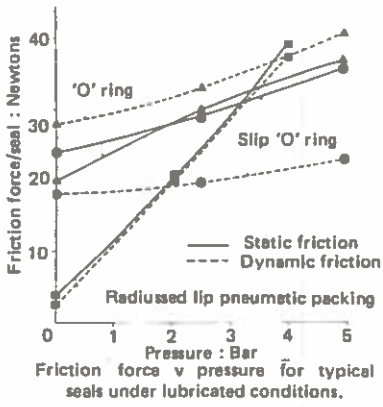
Composite 'hard' rings may also be employed for specific applications, or where favoured by a particular designer. These take the form of a rigid ring section, which provides the rubbing and sealing surface, combined with a pliable cushion ring. The 'hard' section may be of metal or some other low friction material (*eg* PTFE). They are essentially compression seals, with resilience imparted by the cushion section. A possible advantage with rings of this type is that they are capable of acting as bearings as well as seals, with an ability to accommodate side loads and provide an automatic 'self centring' action.

A particular example here is the combination of an O-ring with a rigid PTFE ring, the latter providing a low friction sealing surface irrespective of whether or not lubricant is present. This type of seal can be either an internal or external type (Fig 6) and is produced by various manufacturers under different names (differing in the form of the PTFE ring). Best known in Britain is the (Angus) Slip-O-Ring. A comparison between the performance of this type of composite ring and a conventional O-ring is shown in Figs 7 and 8. Here it is interesting to note that the static



Friction force v pressure for typical seals under dry conditions.

Fig 7



Friction force v pressure for typical seals under lubricated conditions.

Fig 8

friction is similar to that of an O-ring, although its performance under dynamic conditions is superior. Also noteworthy is that at very low pressure, lip type packings show the lowest friction (Fig 7). In this case, however, increasing pressure increases the lip area in contact and substantially increases the friction.

TABLE IA – STANDARD TYPE PISTON SEALS FOR AIR CYLINDERS

Light Duty Cylinders*		Medium Duty Cylinders*		Heavy Duty Cylinders*	
Seal Type	Remarks	Seal Type	Remarks	Seal Type	Remarks
O-rings	Suitable for small cylinders	Cup	Single for single-acting cylinders. Back-to-back for double-acting cylinders.	U-ring	Single or back-to-back.
Cup	Single for single-acting cylinders. Back-to-back for double-acting cylinders	U-ring	As cup above	C-ring	Single or back-to-back
		C-ring	As cup above	V-ring	Chevron set, single or back-to-back
				Solid rings	Automatic type piston rings for high temperature services.

*Proprietary seal designs commonly preferred

TABLE IB – STANDARD TYPE ROD SEALS FOR AIR CYLINDERS

Light Duty Cylinders		Medium Duty Cylinders*		Heavy Duty Cylinders*	
Seal Type	Remarks	Seal Type	Remarks	Seal Type	Remarks
O-ring(s)	Small cylinders	Collar		U-ring(s)	Single or back-to-back
Quad-ring(s)	Small cylinders	U-ring	With or without support ring	C-ring(s)	
Hat	Alternative to collar seal				
Collar	Better resistance to scuffing than hat seal	C-ring	Single or in set	V-ring(s)	Chevron sets, single or back-to-back
U-ring	Larger cylinders			Compression packing	High temperature service and some arduous duties

* Proprietary seal designs commonly preferred

Rod Seals

Largely because of the considerable difference between rod and cylinder diameters, different types of seals may be used on the rod and piston in pneumatic cylinders. Some manufacturers also make available alternative types of seals with a given cylinder design. Even the use of packed glands or similar compression packings for rod seals is not entirely excluded from modern practice, despite their much higher friction than pressure energized seals. There are also applications where relatively high friction on the rod can be an advantage, *eg* to add stiffness to the movement in the case of a vertical air cylinder used to raise a weight.

An additional seal on the outside of the rod seal is also commonly used. This is a wiper seal, the duty of which is to scrape over the entering rod surface and remove dirt, *etc*, which may have collected on it, thus preventing it being drawn into the main rod seal where it could be retained to cause wear. Wiper seals can take the form of simple single lip seals, metallic scraper rings bonded to an elastomeric section, thin elastomeric sections backed by metal to provide support and rigidity, or even simple compression seals, (*eg* felt rings). Polyurethane rubber wiper rings are finding increased favour because of their good wear and abrasion resistance properties. A gaiter may also be added for positive protection where the cylinder is used in particularly dirty conditions, or used in place of a wiper.

Example of superior recovery of 'specialized' pneumatic-duty elastomer.

(a)
Pneumatic packing in H7 material shortly after release

(b)
Pneumatic packing in conventional material shortly after release (strictly comparative time).

Rubbers

Nitrile rubber is the usual choice for elastomeric seals, except when extremes of temperature may be encountered. Both the high temperature and low temperature performance of nitrile rubber can, however, be improved by compounding, although not usually both together. Thus, in order to obtain good low temperature characteristics with a nitrile rubber it is usually necessary to sacrifice some higher temperature characteristic, such as oil resistance.

Hardness values for rubber seal materials may range from soft (40–45°) to very hard (90–95°). Choice of suitable hardness is largely a compromise. A low hardness improves conformability and reduces break-out friction with dynamic seals, and is particularly desirable with low-pressure, pressure-energized seals (*ie* pneumatic seals). A high hardness shows lower rubbing friction and reduces susceptibility to wear, abrasion and extrusion. Rubbing friction, however, will also be dependent on the degree of interference or 'squeeze' present, which for adequate sealing may need to be appreciably greater in the case of a hard rubber than a soft rubber.

A guide to rubber selection for service temperature involved is given in Table II. Temperatures above 150°C (300°F) demand the use of more sophisticated (and more expensive) rubbers, if an elastomeric seal is to be used. All modern rubbers, however, have a generally good low temperature performance down to –30°C or below.

The permeability of elastomers to air is also temperature dependent – see Table III. Here, in fact, it will be seen that silicone rubber is the least favourable material. At normal ambient temperatures the permeability of all other elastomers is low and roughly similar.

TABLE II – WORKING TEMPERATURE RANGE OF TYPICAL ELASTOMERS

Elastomer	MINIMUM °F	MINIMUM °C	MAXIMUM °F	MAXIMUM °C
Butyl rubber	–60	–50	200	95
Ethylene propylene	–65	–54	300	150
Nitrile – *65°	–65	–55	225	110
Nitrile – *70°	–40	–40	200–250	95–120
Nitrile – *80°	–40	–40	225–300	110–150
Nitrile – *90°	–30	–35	300	150
Neoprene	–65	–55	300	150
Polyurethane	–65	–55	300	150
Silicone *60°	–70	–60	450	230
Silicone *70°	–70	–60	450	230
Fluorosilicone	–65	–55	450	230
Viton	–65	–55	450	230

*Shore hardness

TABLE III – AIR PERMEABILITY OF ELASTOMERS
(Permeability coefficient Q x 10^8 cm^2/sec/bar)

Elastomer	Temperature °C 40	60	80	100	120	140	160	180
Acrylic	5	13	30	38	45	60	75	93
Butyl	0.5	1.8	4.5	10	19	28	40	60
Neoprene (Cr)	2	4.5	10	17	26	38	52	71
Nitrile	2	4.5	8	15	22	31	46	63
SBR	3.5	9	18	30	45	70	100	115
Natural rubber	9.5	20	4.3	56	70	100	110	120
Silicone	290	380	475	580	660	755	900	1000
Fluorocarbon	1	3	9	20	36	60	90	150
Polyurethane	2	4.5	10	19	31	40	52	70

Pressure Gauges

COMPRESSED AIR systems normally operate within a pressure range up to 10 bar (140 lb/in^2). Where indication of pressure is required a Bourdon gauge is the usual choice, a typical construction being shown in Fig 1.

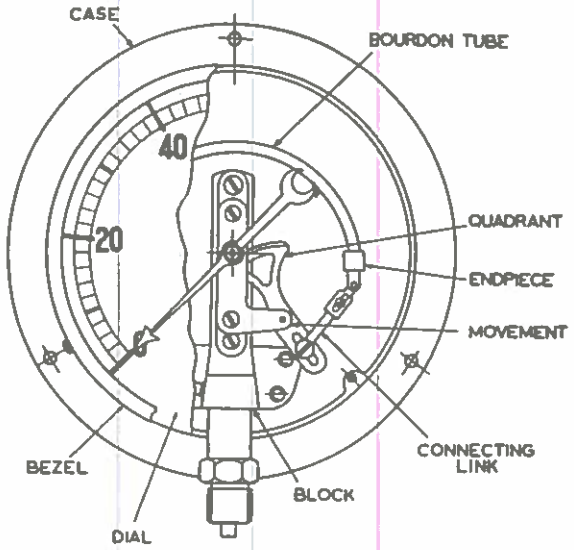


Fig 1 Typical arrangement of Bourdon gauge.

The basic element is a thin walled tube, slightly flattened in section and bent into a 'C' shape. One end is fixed and the free end is sealed and attached to a mechanical linkage. Application of internal pressure to the tube causes it to straighten by an amount directly proportional to the internal pressure. The resulting movement of the free end is transmitted via the mechanical linkage to rotate a pointer associated with a graduated dial. This mechanical movement involves multiplication so that something like 300° of pointer rotation is achieved over the working pressure range of the gauge.

Accuracy achieved with a Bourdon tube gauge can be of the order of plus or minus 2½% of the full scale deflection. Greatest accuracy is normally achieved at the upper end of the scale. Thus selection of a Bourdon tube gauge should be based on pressure range where the normal reading point will be toward the upper end of the scale. Consistency of performance, however, is related to the fatigue characteristics of the tube material and some drift is inevitable with time. As a general rule, therefore, Bourdon gauges need regular checking and recalibration as necessary when in use.

Bourdon gauges are susceptible to mechanical shock and pressure surges and should be protected against both forms of disturbance. A simple method of protection against pressure surges is to fit a snubber or restrictor in the feed line. In severe cases, mechanical throttling, or a shut-off-device, responding directly to any sudden increase in pressure, may be employed.

Another possible source of trouble is blocking of the tube by contaminants in the fluid. To obviate this a diaphragm type seal can be incorporated in the inlet or feed. The Bourdon tube is then filled with clean fluid (usually oil) which is isolated from the working fluid, the pressure of the latter being transferred to the Bourdon tube through the flexible seal. This principle can be extended by employing a diaphragm unit or capsule as a transmitter, separate from the Bourdon gauge and connected to it by a length of tubing. A distinct advantage offered here is that the gauge itself can be distant from the transmitter without affecting the accuracy of the reading.

Modifications to the original form of element include the use of helical tubes, some examples being shown in Fig 2.

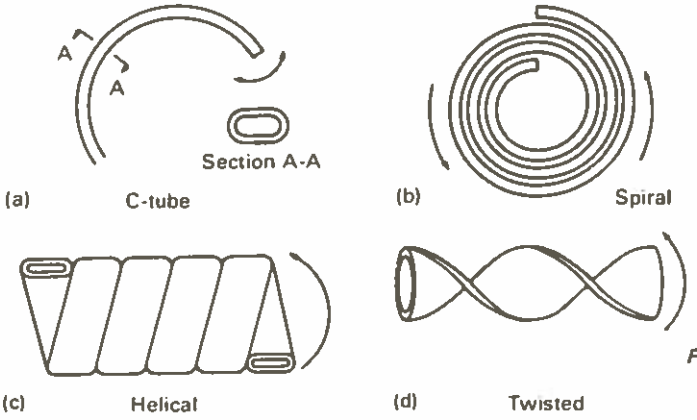


Fig 2 Bourdon tubes (Schaevitz)

The spiral tube is similar to the C-tube, except that its radius of curvature changes along the length of the tube, creating a greater length of tube for a given circumference. The number of turns in the spiral vary. Because the active length of tube is larger, the arc length of stroke is greater. A straight-line pickoff from this larger angle of sweep causes a larger non-linearity in output displacement than with a C-tube.

The helical tube permits a long tube length for a given radius of curvature. The average tube has from two to six turns. The movement of the tip is greater than that for the C-tube and is more nearly circular. The increase in length of the helical tube while pressurized is negligible. The travel of the tip lies in a plane perpendicular to the centre line of the helix, increasing as the active tube length increases. The tip travel is in the same order of magnitude (per unit length of tube) as an equivalent C-tube.

The twisted tube maintains a straight centre line along the length of the tube but is twisted about the centre line at a uniform rate. The expansion of the tube by internal pressure causes the tube to unwind angularly about the tube centre line. The angle of rotation is proportional to applied pressure. The change in tube length when the tube unwinds is negligible. Different angles of twist alter the angular rotation for any given pressure and length of tube.

Electronic Pressure Gauges

Electronic pressure gauges employ a pressure transducer normally connected into a resistance bridge network, followed by an amplifier. Pressure may be indicated by a meter or, in modern designs, through a complementary circuit with digital readout via light emitting diodes. The pressure range depends on the type of transducer used, but the general range covered by instruments of this type is 0–700 bar (0–10000 lb/in^2). Accuracy again depends mainly on the transducer and normally compares favourably with, or is better than, a comparable Bourdon gauge. Accuracy is normally maintained to within ± 1% of the maximum pressure rating of the transducer.

Diaphragm Pressure Gauges

Simple diaphragm pressure gauges are suitable only for the measurement of relatively low pressures, *eg* typically up to 1 bar (14.5 lb/in^2). They are, however, particularly sensitive devices and well suited to the measurement or detection of small differential pressures fed to either side of the diaphragm. A simple diaphragm gauge of this type is shown in Fig 3. The travel of the diaphragm, and thus the stroke available to operate an indicating movement, is dependent on the cross sectional form of the diaphragm and its diameter. Extreme movement may only be obtained at the expense of loss of linearity of movement.

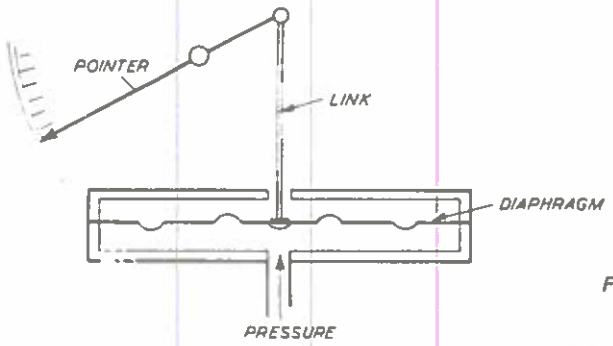


Fig 3

The stroke of a single diaphragm gauge can be extended considerably by employing a slack diaphragm, made from a suitable limp material such as nylon, oiled silk or thin leather. Such a diaphragm requires biasing with spring pressure opposing the direction of applied pressure (or of the highest applied pressure) and stops to limit the movement. The particular advantage offered by a slack diaphragm is the large diaphragm area which can be exposed to pressure, making the unit particularly sensitive for the measurement of very low pressures.

Metallic diaphragms offer the possibility of working at higher pressure ratings, although the movement of stroke is largely dependent on the extent of corrugation, for a given diameter. Performance can generally be improved, both as regards stroke and pressure sensitivity, by fabricating a series of diaphragms in the form of a bellows, although here the contour adopted can have a bearing on linearity of stroke, and vice versa.

Welded metallic diaphragm gauges, based on a built-up bellows shape or series of capsules, are mostly used for pressure measurements within the range 25 mm wg to 0.35 bar (1 inch wg to 5 lb/in^2). In the case of capsule types, the capsules form hermetically sealed chambers which may be partially evacuated to render the element more sensitive — *eg* the type of element used in an aneroid barometer.

Bellows type elements formed from thin-walled tubing have a higher pressure rating, and such elements may be used in several ways. For pressure measurement within the range of 0.35–2.50 bar (4–50 lb/in^2), or vacuum pressures up to 762 mm Hg (30 in Hg) an open ended bellows is commonly used, enclosed within a chamber. Pressure is then applied to the chamber, with mechanical movement derived from the free end of the bellows. This basic layout may be further modified by the application of spring loading, opposing the movement produced by the pressure force. This expands the bellows where the applied pressure to be measured is positive, and contracts the bellows in the case of measurement of vacuum. (See also Fig 4).

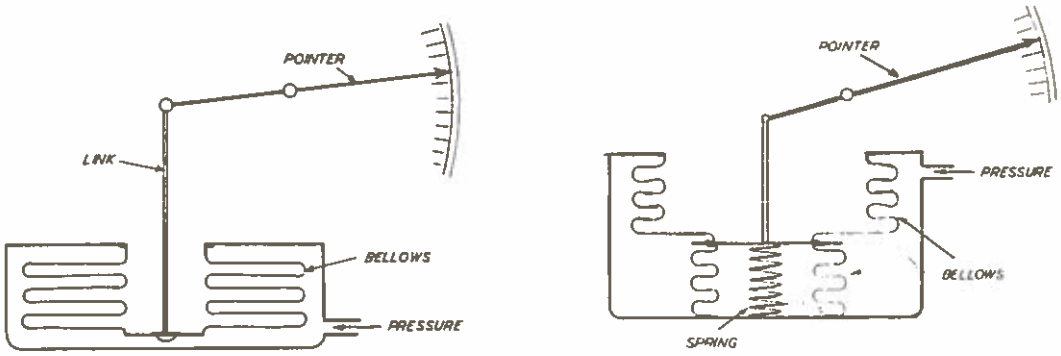


Fig 4

Sensitivity may be improved by the use of additional bellows to apply spring bias. Alternatively, matched bellows may be used, each in a separate sealed chamber. The interiors of the two bellows are then interconnected and filled with a liquid with a high bulk modulus, such as oil or water. Pressure is applied to one chamber, and causes a contrary movement in the other bellows due to liquid displacement, thus doubling the mechanical movement available from the applied pressure. Alternatively, if one of a pair of matched bellows is evacuated and sealed, and the other is connected to the pressure source, the movement will read absolute pressure directly.

Vacuum Gauges

Conventional pressure gauges of the capsule or bellows type may be suitable for pressure measurements down to 25 torr. This pressure range is generally referred to as low vacuum. For higher vacua, *ie* lower absolute pressures, special vacuum gauges are employed.

The Mcleod gauge is a special form of mercury manometer with three limbs, one of which is sealed at the top. Pressure is determined by measurement of the degree of compression of the gas trapped in the sealed limb. The normal pressure range of such a gauge is from 10 torr down to 10^{-4} torr. It is more suitable for laboratory work rather than as an industrial instrument, but more robust and easier to use gauges working on a similar principle have been developed for general use.

The Pirani gauge is based on the behaviour of a heated wire element enclosed within a suitable gas chamber and connected to a Wheatstone bridge circuit. The principle involved is that the

thermal conductivity of the gas, and thus the temperature of the wire supplied with a constant heater voltage, is pressure dependent. This change is indicated as a change in resistance of the wire element, measured on the bridge circuit. Correction for ambient temperature can, if necessary, be arranged by installing a second gauge in the opposite arm of the bridge. Normal range of a Pirani gauge is from 5 torr down to about 10^{-4} torr.

Thermocouple gauges have also proved suitable for pressure measurement in the range 10^{-1} to 10^{-4} torr. These work on a similar principle to the Pirani gauge, except that the temperature of the element is measured directly (by thermocouple reaction) rather than indirectly as a change in resistance.

For measurement in the higher ranges ionic or radionic gauges are normally employed. These are specialized devices giving meticulous calibration and control (*eg* the use of stabilized circuits). Degassing can also be a problem, upsetting the calibration and sensitivity of such gauges at high vacua. Measurement range is, however, readily extended down to 10^{-6} or 10^{-7} torr.

Gauge Pressure

Pressure-measuring devices normally measure the difference between applied pressure and atmospheric pressure and thus indicate *gauge pressure* (or vacuum pressure in the case of a vacuum gauge). The equivalent *absolute* pressure is then given by relating this reading (gauge pressure) to zero pressure as a datum – see Fig 5. Thus in the case of positive pressure (above atmosphere) absolute pressure is the sum of the gauge pressure and atmospheric pressure (normally assumed to be 1 bar or 14.5 lb/in^2).

Since pneumatics is normally concerned with gauge pressures of a relatively low numerical order (*eg* less than 10 bar), the difference between absolute pressure and gauge pressure can be significant.

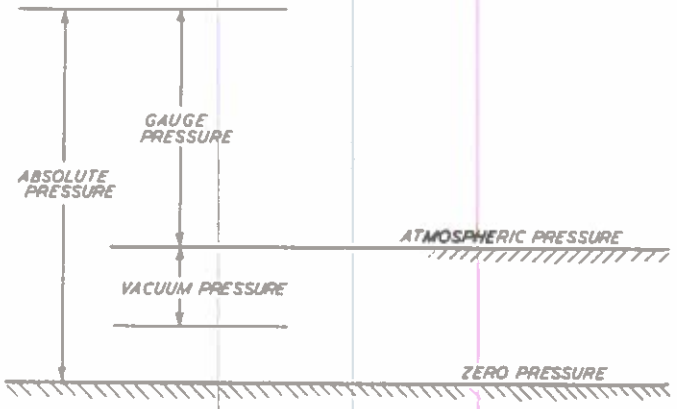


Fig 5

Pressure Units

The standard pressure unit employed in pneumatics is the *bar* or 10^5 N/m^2. This is slightly different from the original metric pressure unit kg/cm^2 although this difference can be ignored for most practical purposes.

1 bar = 1.02 kg/cm^2 = 14.5 lb/in^2
1 kg/cm^2 = 0.98 bar = 14.2 lb/in^2

The definition of a *standard atmosphere* (atm) remains as 1 013 250 dyn/cm^2, whereas 1 bar = 10^6 dyn/cm^2.

Hence

1 atm = 1.013250 bar
= 14.695975 lb/in^2
= 1.0332299 kg/cm^2

The *technical* atmosphere (at) is also still used in Continental Europe, equivalent to a standard atmosphere pressure of 1.0 kilopond per cm^2 (kp/cm^2).

Here approximate equivalents are:-

1 at = 1.0 kp/cm^2
= 0.998 kg/cm^2
= 0.978 bar
= 14.2 lb/in^2
1 bar = 1.022 at
1 atm = 0.965 at

Differences can thus arise in numerical values of compression ratios calculated on the basis of the bar being adopted as the nominal equivalent of standard atmospheric pressure, compared with the use of atm or at.

The most convenient (and generally accepted) pressure unit for vacuum measurement is the *torr,* corresponding to an absolute pressure of 1 millimetre of mercury. Normal atmospheric pressure is 760 torr. Various other units may be quoted, however, *eg*

Microns or Micromillimetres — where 1 micron equals 10^{-3} torr

Millibars — where 1 torr equals 1.33 millibars, or 1 millibar equals 0.75 torr

Microbars — equivalent to dyn/cm^2 or millibars x 10^{-3}

Per cent vacuum — which may be defined as —

$$\frac{760 - \text{pressure in torrs}}{7.6}$$

Empirical units — such as mm of mercury (mm Hg), inches of mercury (in Hg), inches of water, *etc.*

Air Flow Measurement

METHODS OF air flow measurement are detailed in BS1571:1975 Part 1. The method normally employed is testing with a standard nozzle on the delivery side together with a perforated plate at the inlet flange of the approach pipe to smooth the flow — Fig 1. A receiver of adequate size must be fitted between the compressor and control valve to dampen any pulsations and thus ensure accurate flow measurement.

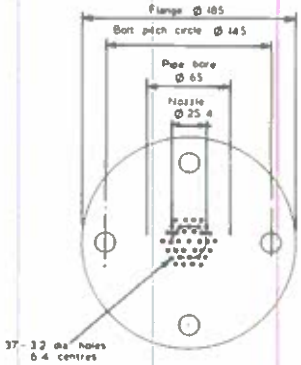


Fig 1 BS perforated plate for flow smoothing.

Where the pressure drop across the control valve is substantial (not less than 30% of the upstream absolute pressure), an adequate capacity receiver is one which would contain, at the receiver pressure, the output of the compressor or exhauster during 50 complete pulsations.

Where the pressure drop across the control valve is less than 30% of the upstream absolute pressure or where no control valve is used between the receiver and the nozzle, a larger receiver may be necessary.

Other damping arrangements may be used provided that it can be shown that no pulsations exceeding ±2% of the differential pressure are occurring at the nozzle inlet.

The pipe connecting the machine and the receiver should be as short as possible to minimize the risk of false readings due to resonance effect.

Testing may be done with the nozzle on the delivery side, or the intake — see chapter on *Compressor Acceptance Tests.* In both cases pipes and nozzle are circular in section. Pipe flanges

are preferably faced so that a metal-to-metal joint is made with the nozzle. If jointing material is used between the flanges and the nozzle its thickness must not exceed 1.6 mm and the inner diameter of the joint ring must be not less than the appropriate value of dimension G given in Table I.

Testing with Nozzle on the Delivery Side

The control valve may be either the same nominal size as the nozzle approach pipe or smaller. The latter may be preferred for ease of control, and it should then be connected to the nozzle approach pipe by a cone piece. In either case a perforated plate must be fitted at the inlet flange of the approach pipe to smooth the flow, which leaves the control valve in a violently turbulent condition. The perforated plate must have an hexagonal pattern of holes drilled in the central portion, the centres of the holes being uniformly spaced and located at the corners of equilateral triangles. The number, size and spacing of the holes, and thickness of the plate are specified in Table II.

The pressure drop across this perforated plate is about four times the nozzle head, hence if it is desired to test up to the maximum nozzle head of 1 000 mm H_2O, the minimum pressure needed would be about 0.5 bar gauge if discharging to atmosphere. If tests are to be made at lower supply pressures, the nozzle size can be chosen to work at a lower head.

Generally the air pressure on the outlet side of the nozzle is only slightly in excess of atmospheric pressure because the exit pipe discharges either freely to atmosphere or through an exhaust or silencing system to atmosphere.

Downstream from the perforated plate, the approach pipe to the nozzle must be straight with a length not less than 8 pipe diameters. The pipe bore must not be greater than 2.5 nor less than 2.25 nozzle diameters except for the two smallest nozzle sizes in Table I (where it must be 25.4 ±3.2 mm).

The exit pipeline from the nozzle to atmosphere must be straight for a distance not less than 4 pipe diameters and of the same diameter as the approach pipe.

The roughness of the internal surface of the pipes need not be reduced below that of ordinary commercial finish, but there must be no obstructions in the pipelines such as thick welding ridges.

Testing with a Nozzle on the Intake

When the nozzle is on the intake the approach pipe to the nozzle must have a minimum straight length of 8 pipe diameters and the exit pipe a minimum straight length of 5 pipe diameters. If aspiring from atmosphere, the inlet end may be plain, flanged or flared, but must not be restricted. As the control valve is on the downstream side of the nozzle a perforated plate is not necessary.

If drawing from another pipe system, special precautions should be taken to avoid swirl or other flow disturbance; *eg* by using an appropriate perforated plate at the entry to the straight pipe of 8 pipe diameters length.

Nozzle Selection

The following empirical rule can be used to ascertain a size of nozzle suitable for a particular test, when the pressure at the nozzle is approximately atmospheric.

Calculate 0.25 times the maximum expected volume flow in litres/s and select the nozzle with a nozzle constant (k) next larger than that number. Nozzle constants for BS rounded nozzles are given in Table III.

For other pressures the approximate value of k may be found from the formula for Q (see later) using a value of h within the range given in Table III.

TABLE I – DIMENSIONS OF NOZZLES (BS1571:1975)

Nominal Size												
Nozzle diameter *d*		Pipe bore *D*	*d*	*A*	*B*	*C*	*E*	*F*	*G*	*H*	*J*	*K*
in												
7/32	5.56	25	5.56±0.01	3.18±0.06	—	—	—	19.05	44.45	—	0.40	—
3/8	9.5	25	9.53±0.01	5.77±0.06	2.87	3.15	1.91	19.05	44.45	—	0.40	—
5/8	15.9	40	15.88±0.01	9.60±0.06	4.78	5.23	3.18	26.19	60.33	—	0.40	—
1	25.4	65	25.4±0.03	15.37±0.08	7.62	8.38	5.08	38.89	85.73	11.11	0.80	29.77
1½	38.1	90	38.1±0.03	23.06±0.10	10.29	12.57	7.62	51.59	111.13	11.11	1.60	44.45
2½	63.5	150	63.5±0.05	38.43±0.13	19.05	20.96	12.70	84.93	177.80	11.11	2.40	74.22
4	101.6	270	101.6±0.05	61.47±0.13	30.48	33.53	20.32	141.29	292.10	12.70	3.20	117.67
6	152.4	375	152.4±0.10	92.20±0.18	45.72	50.29	30.48	206.38	425.45	15.88	4.80	177.80
10	254.0	600	254.0±0.18	153.67±0.25	76.20	83.82	50.80	323.85	660.40	19.05	8.00	293.10
15	381.0	900	381.0±0.25	230.51±0.38	114.30	125.73	76.20	497.43	971.55	19.05	12.20	419.10

All dimensions are in millimetres

TABLE II – DETAILS OF PERFORATED PLATE (BS1571:1975)

Nominal nozzle diameter *d*		Nominal pipe bore *D*	Details of perforated plate			Thickness of plate
			Number of holes	Diameter of holes	Spacing of holes	
in	mm	mm		mm	mm	mm
7/32	5.56	25	19	1.6	3.2	3.2
3/8	9.53	25	19	1.6	3.2	3.2
5/8	15.88	40	61	1.6	3.2	3.2
1	25.4	65	37	3.2	6.4	3.2
1½	38.1	90	61	3.2	6.4	3.2
2½	63.5	150	217	3.2	6.4	3.2
4	101.6	250	169	6.4	12.7	6.4
6	152.4	375	331	6.4	12.7	6.4
10	254.0	600	217	12.7	25.4	12.7
15	381.0	900	469	12.7	25.4	12.7

TABLE III – ROUNDED NOZZLE: NOZZLE CONSTANTS AND VOLUME FLOW (BS1571:1975)

Nominal nozzle diameter *d*		Approximate pipe bore *D*	Constant *k*	Approximate volume flow for nozzle outlet pressure near atmospheric			
				h = 10 mm H_2O	h = 100 mm H_2O	h = 400 mm H_2O	h = 1000 mm H_2O
in	mm	mm		litres/s	litres/s	litres/s	litres/s
7/32	5.56	25	0.1488	—	1	2	3
3/8	9.53	25	0.4432	—	3	6	8
5/8	15.88	40	1.254	—	8	16	24
1	25.4	65	3.248	—	20	40	60
1½	38.1	90	7.377	14	43	85	140
2½	63.5	150	20.516	38	125	250	380
4	101.6	250	52.429	100	310	620	1 000
6	152.4	375	118.41	220	700	1400	2 200
10	254	600	328.00	560	1900	3900	5 600
15	381	900	737.68	1450	4400	8700	14 500

Observations and Measurements

The following observations are necessary to conform to British Standard practice.

(i) *Atmospheric pressure*

Measured in mm Hg by means of a barometer. If there is no barometer on site, readings can be obtained from a local recognized weather bureau and corrections applied to allow for the difference in altitude.

(ii) *At the agreed inlet point*

The difference in pressure between the agreed inlet point and the local barometer reading. For the purposes of calculation the pressure at the inlet point is the barometer pressure less the depression at the agreed inlet point.

The temperature of the air at the agreed point in °C. The thermometer shall be so placed that it is not affected by radiant heat from the surroundings.

(iii) *At the receiver*

The receiver pressure in bar measured by means of a pressure gauge. This is taken as a record of performance but is not used in the calculation of the flow except where necessary to allow for any small change occurring during the test. Such changes shall not exceed 2% of the absolute pressure in the receiver.

(iv) *At the nozzle*

The head across the nozzle, measured as the difference in pressure given by the reading of a double-leg manometer.

The difference in pressure between the downstream side of the nozzle and the atmosphere. This shall be measured by means of a second manometer connected as shown in Fig 1.

The temperature in the pipeline, measured by means of a thermometer as shown in Fig 2. This thermometer shall be placed between three and four pipe diameters downstream from the mouth of the nozzle, and not closer than one pipe diameter to an open end.

Instruments used for these and other measurements are summarized in Table IV.

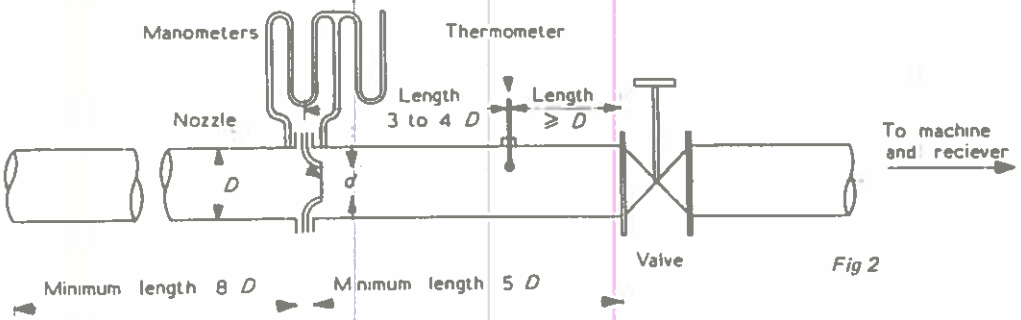


Fig 2

TABLE IV – MEASURING EQUIPMENT AND METHODS
(As specified in BS1571:Part 2)

Quantity	Instrument(s)	Exclusions	Precautions
Temperature	Mercury in glass thermometers with etched stems (or other methods as given in BS1041)	Conventional or industrial metal-cased thermometers	Air velocity not to exceed 30 m/sec at point of measurement
Pressure (above 2 bar)	Calibrated Bourdon gauge, dead-weight gauge, mercury barometer	Diaphragm gauges	Pressure readings to be between ¼ and ¾ full scale using Bourdon gauges
Pressure (below 2 bar)	Manometer or columns (or vacuum gauge)	Closed mercury columns for vacuum measurement	Eliminate pulsating flow (eg with receiver)
Air Flow	B.S. nozzle		See text
Shaft speed	Tachometer or speed counter		Instrument should be checked regularly for accuracy
Power	Electric motor efficiency (BS2613), or dynamometer		Calculation of output based on average of increasing and decreasing loads

Method of Calculation (BS1571:Part 2)

The capacity Q, expressed in litres/s, is calculated from the formula:

$$Q = k. \frac{T_1}{p_1} h. \frac{p_2}{T}$$

where Q is the inlet volume in litres/s;
h is the pressure drop across the nozzle in mm H_2O;
k is the constant for the nozzle;
T_1 is the absolute temperature at the agreed inlet point;
T is the absolute temperature downstream from the nozzle;
p_1 is the absolute pressure at the agreed inlet point in mm Hg;
p_2 is the absolute pressure at the downstream side of the nozzle in mm Hg.

These absolute pressures are obtained by adding or subtracting from the atmospheric pressure, as given by the barometer, the water-gauge reading obtained from the appropriate manometer. The water-gauge readings are converted to mm Hg by dividing by 13.6.

Alternative Tests

The discharge may have to be made into a pipeline under pressure instead of into the atmosphere, or it may be desirable to measure the flow at the delivery pressure in order to reduce the dimensions of the pipeline required. The test layout should then be the same as that shown in Fig 3 with the addition of a second perforated plate across the downstream pipe at least five pipe diameters away from the nozzle and before the flow reaches any bends, valves or obstructions. The pressure at the downstream side of the nozzle is measured by a pressure gauge substituted for the second manometer and the reading converted to mm Hg.

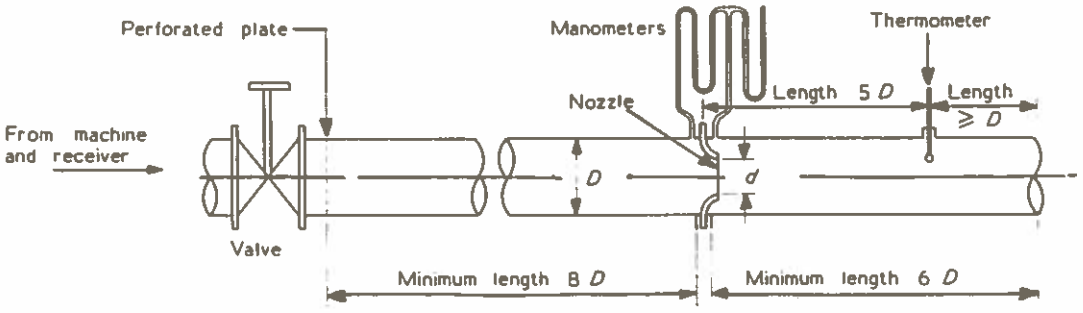


Fig 3 Arrangement for testing with a nozzle on the delivery side.

$$2.25 < D < 2.5$$

Bourdon gauges are mounted vertically and must be readable to 0.5% of the full scale reading. They must be used only between one quarter and three quarters of their full scale reading. They must be calibrated against a dead-weight gauge before and after the test.

Care must be taken to ensure that the high pressure pipeline into which the air is discharged is free from pulsations.

The calculations are made in the same manner as for discharge to atmosphere.

SECTION 3

Circuits

Pneumatic Circuits

A PNEUMATIC circuit is normally distinguished by being open-ended, *ie* the compressed air is exhausted to the atmosphere after passing through the actuator. In practice exhaust facilities are usually provided by the control valve or valves employed in the circuit, requiring only a single line connection between valve and actuator. It is also a general character of such circuits that no power is consumed in a system under pressure with no motion, unless there is a deliberate bleed or accidental leakage. Hence depressurization is not necessarily required to save power after a movement.

Actuators normally comprise air cylinders where linear motions are required, or air motors for rotary motions and drives. The variety of circuits which can be derived in the former case is virtually infinite, ranging from simple stroking motions with manual control to fully automatic working allied to sophisticated automatic or programmed control systems.

Single-Acting Cylinders

Generally a 3-way, normally closed valve is used to control a single-acting cylinder, as shown in pictorial form in Fig 1. In this diagram port no.1 is the inlet, port no.2 the outlet and port no.3 the exhaust. Fig 1a then shows the valve with the spool in the normal position with the inlet

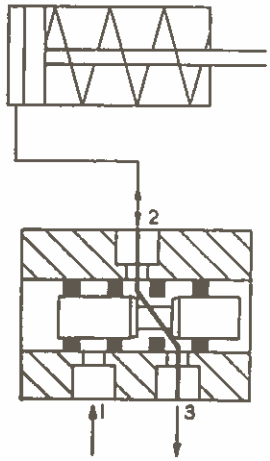


Fig 1a

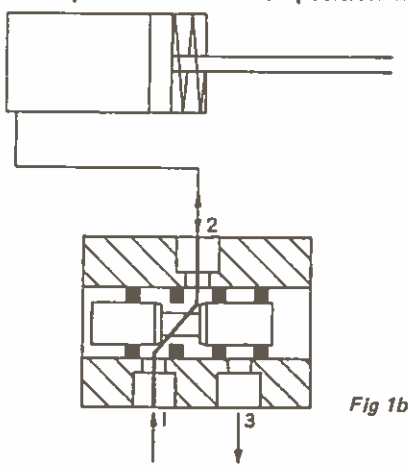


Fig 1b

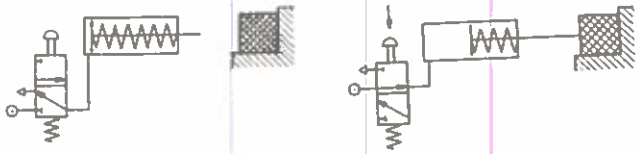


Fig 2

closed and outlet open to exhaust. Fig 1b shows the valve in the operating position where inlet is open through to outlet and the exhaust is closed.

This circuit is shown in symbol form in Fig 2. Note that circuits are always drawn so that they show the neutral or 'at rest' position of the various circuit elements, *ie* valves, in their normal (unoperated) position and actuators in their unactuated position. This particular circuit diagram shows a manually operated valve with two stable positions. If the valve was fitted with a spring for mechanical return, the cylinder would return to its original position as soon as normal actuation of the valve ceased.

Double-Acting Cylinders

Double-acting cylinders are usually controlled by a 4-way, 5 port two position valve — shown in pictorial form in Fig 3. This valve has one inlet port (1), two cylinder ports (2 and 3), and two exhaust ports (4 and 5). In the first position of the valve (Fig 3a), supply is directed to cylinder port no.2, with cylinder port 3 connected to exhaust, port 5. In the second position supply is connected to cylinder port 3 with cylinder port 3 connected to exhaust port 4.

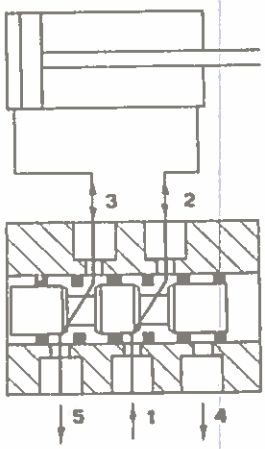


Fig 3a

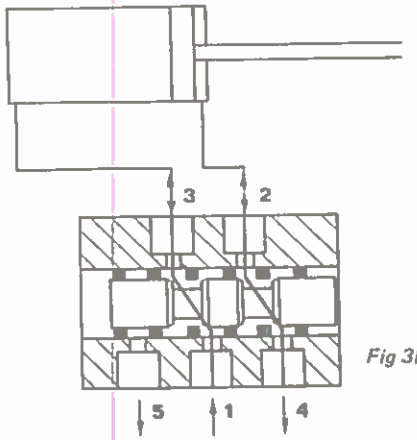


Fig 3b

This circuit is shown in symbolic form in Fig 4. Fig 5 shows this circuit extended to embrace speed control with a restrictor placed prior to the valve in the supply line, and Fig 6 with restrictors in the outlet. The latter has the advantage that speed can be regulated in either direction independently. Again, in practice, a better solution is usually to regulate the exhaust air from the cylinder with one-way restriction valves, as in Fig 7.

Automatic return in the case of a manually operated control valve can be initiated by spring return of the valve. If the control valve is pilot operated (*ie* from a separate pilot valve), automatic return can be provided by an additional mechanically operated return valve — as in Fig 8.

To extend cylinder operation to automatic reciprocation, an additional pilot valve is needed, as shown in Fig 9. Using a lever-operated start-stop valve, this has a further provision that if this valve

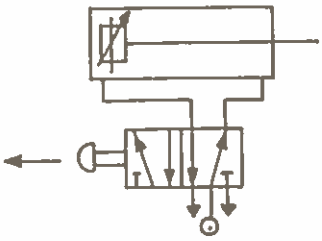


Fig 4

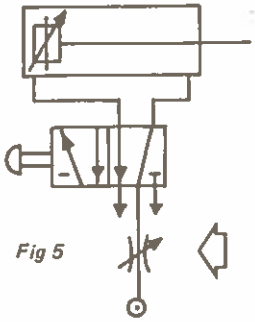


Fig 5

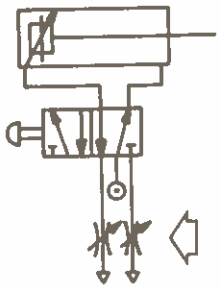


Fig 6

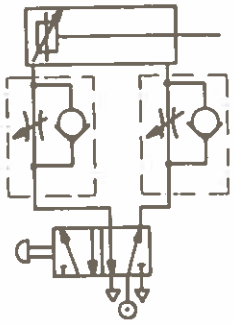


Fig 7

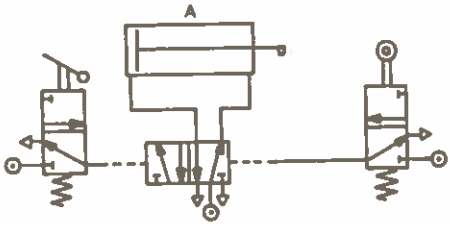


Fig 8

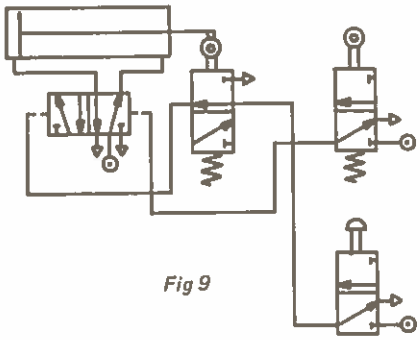


Fig 9

is returned to stop, the supply is cut off and the piston stops in the extended position. With the start-stop valve left in the 'on' position, the cylinder will continue to reciprocate.

Numerous other variations are possible on this type of circuitry. Mechanically tripped valves offer simplification of the circuit since they need no separate controlling air supply.

Two faults can readily arise in the design of more complicated pneumatic circuits. The first is 'over design' where the system is unnecessarily complicated, or incorporates an excessive number of components (some of which may actually be redundant when considered in paired action). If the problem is re-thought, rather than the original concept pursued, resulting in increasing complexity, a very much simpler solution will often emerge. The second is the possibility of the paper design producing a locked circuit, or definite circuit fault, by omitting to appreciate that a signal can remain uncancelled in a particular part of the circuit, with locking effect. Ironically, this often arises from a determination to avoid the fault of 'over design', and can be worse in the final result. An 'over designed' circuit will perform the functions required, but an 'over simplified' circuit may not work in practice.

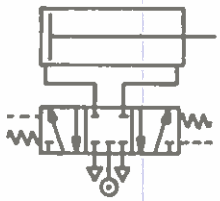


Fig 10

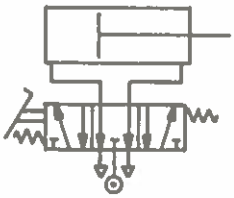


Fig 11

Three-Position Valves

Three-position valves can be used either to provide 'locking' or 'relieving' of a cylinder. A basic 'locking' circuit is shown in Fig 10, and a basic 'relieving' circuit in Fig 11.

'Two-hand Release'

This description is given to the type of circuit used in presses and similar heavy-duty actuators to eliminate the risk of starting when the operator has his hands in a particularly dangerous position. To eliminate such risk, the circuit is designed so that actuation is only possible when both hands are placed on control valves.

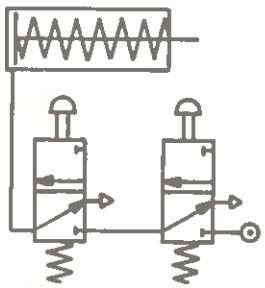


Fig 12

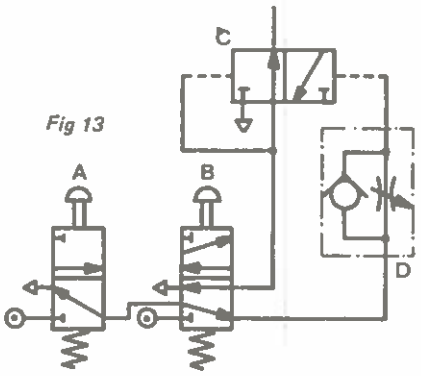


Fig 13

The basic circuit shown in Fig 12 is not a satisfactory solution since the circuit is capable of single handed actuation if one of the valves is pneumatically actuated. The actuator circuit shown in Fig 13 is 100% safe, working as follows:-

If only A is actuated, the air will go to C via B and D at which D switches, which only causes the out-signal to be bled in C and not through B as before. If only B is actuated, the air goes to C via D at which C switches, and the out-signal is bled through C and not through B—A. If A and B are actuated simultaneously, the two pipes to C are aired, but due to the restriction D the signal from A—B will always be first and the out-signal is a fact.

The out-signal can be directly connected to a single-acting cylinder. If a double-acting cylinder is to be controlled, the out-signal has to be connected to a 5-port working valve with mechanical spring return (monostable operating valve) that controls the double-acting cylinder.

Stop Devices

In Fig 9 the working unit will stop in its minus position when the control valve is closed since the position transmitting valve will then not receive any air. If the control valve were instead connected to the position transmitting valve in the plus position, the working cycle would stop in that position.

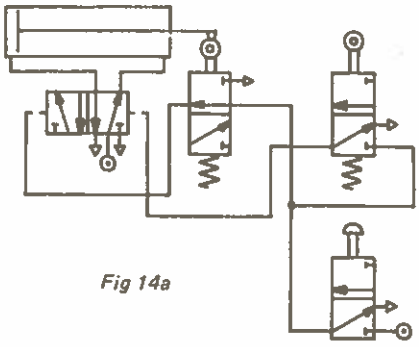


Fig 14a

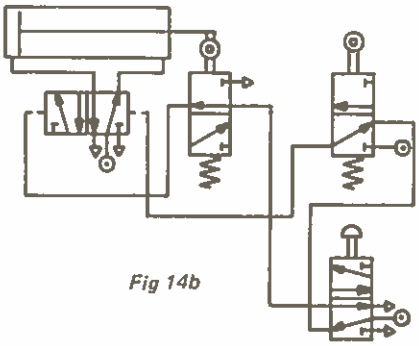


Fig 14b

If the unit is only to finish the movement started when the stop signal is given, the air to both the position transmitting valves has to pass through the control valve as per the diagram in Fig 14a.

If the cylinder is to adopt its minus position regardless if it moves plus or minus, the connection is to be as shown in Fig 14b.

If the cylinder is to stop instantly, the inlet air to the working valve can be cut off, but this brings the following disadvantages:

No control of the position in which the cylinder stops.

No force on the cylinder.

No control of the movement when the process is started again.

These disadvantages can be eliminated if the connection is done according to Fig 15.

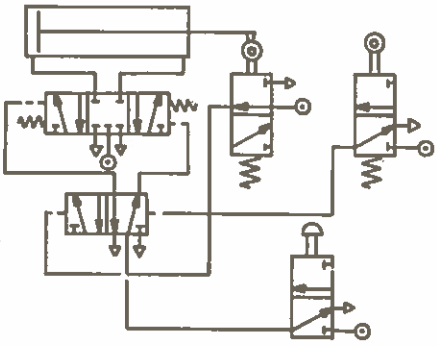


Fig 15

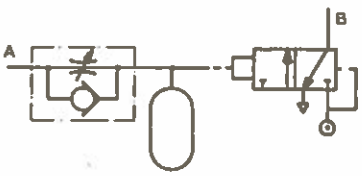


Fig 16

Time Delay

If a cylinder, for example, is to stop in its plus position for a certain period of time before it returns, or if, for construction reasons, a position transmitting valve cannot be placed at the plus position, a time-delaying circuit can be used. Such a circuit is obtained by means of a one-way-restriction valve, an air reservoir and a differential valve Fig 16.

The circuit functions so that when air is connected (in-signal) owing to the restriction valve, a certain time will pass before the pressure in the reservoir becomes so high that the differential valve switches. The delayed out-signal is then received. When the in-signal ceases, the differential valve is reset, as the air in the reservoir is exhausted via the check valve. Time delays from 0 to 5 minutes can be obtained with this type of circuit by adjusting the restriction valve and the volume of the reservoir.

An alternative to the time circuit shown in Fig 16 is to regulate the pressure on the smallest area of the differential valve which gives one further setting possibility Fig 17.

The circuit diagram Fig 18 shows a cylinder that remains in its plus position for a certain time because of the time circuit that delays the acknowledgment signal from the position transmitting valve in the plus position. And the connection diagram, Fig 19 shows a cylinder that returns automatically after a certain time, after having received a signal to go plus.

The connection system works as follows:

When the control valve is actuated, air runs to the working valve that switches and the air flows to the plus chamber of the cylinder. At the same time the air flows to the time circuit which, after a certain time, makes the working valve switch and the cylinder goes minus. In order to obtain a distinct switch of the working valve, the system is provided with a back feed of the air for the plus movement of the cylinder via the control valve to the working valve. This back feed makes it necessary for the working valve to be of differential type as can be seen from the Fig 19.

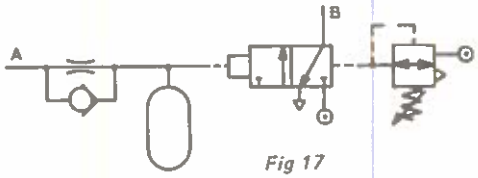


Fig 17

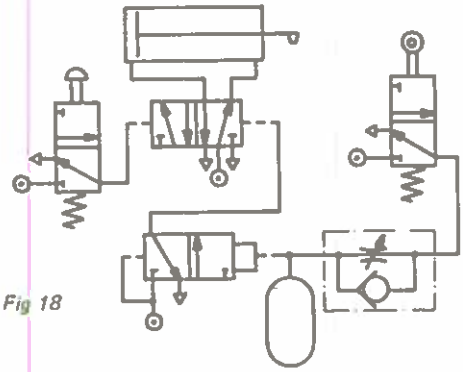


Fig 18

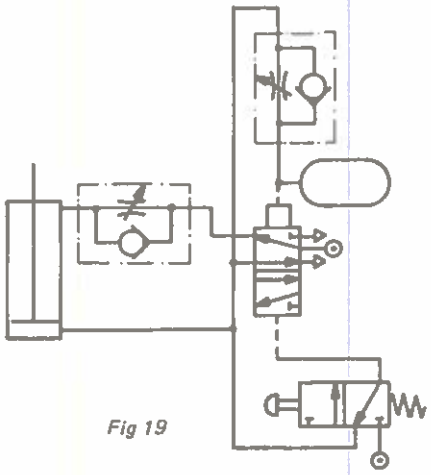


Fig 19

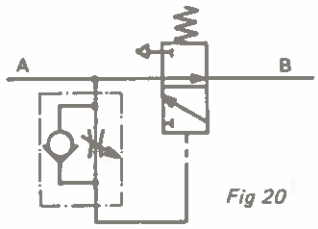


Fig 20

Pulse Circuit

There is often a demand for a signal to be cut out after a certain time, regardless of the duration of the in-signal, *eg* for cleaning. This is achieved in the way shown in Fig 20.

When the in-signal A comes, it goes directly through the valve which is of 'normally open' type. At the same time, air flows via the restriction valve to the actuating part of the valve. Depending on the restriction and the volume after the restriction valve (normally no extra volume

is needed, as the volume of the conduit is sufficient), the valve switches after a certain time and the out-signal B ceases, whether or not the in-signal A remains. When A disappears, the signal conduit is bled through the check valve.

Control by Pressure Drop

In Fig 19 an example was shown of how to make a cylinder return without using a position transmitting valve in the plus position. Another way of obtaining return without a position valve is to sense the pressure in the minus chamber of the cylinder. This is done by inserting a 3-port differential valve, the larger area of which is connected to the de-airing minus chamber Fig 21.

When the minus chamber is under pressure *ie* during the minus movement, and when the piston is in its minus position, there is an overpressure in the minus chamber. But, also during the plus movement, there is a certain overpressure in the minus chamber, and consequently the differential valve will also be actuated during the plus stroke.

To avoid the pressure in the minus chamber decreasing too fast and thus returning the cylinder before it has reached the plus position, a restriction valve has to be connected to the cylinder connection where the pressure is indicated. When the piston reaches its plus position, or continued movement is otherwise obstructed, the pressure in the minus chamber decreases and the differential valve switches, as does the 5-port valve which controls the cylinder that goes minus. The pressure indication described can, of course, be used in both cylinder directions.

In this kind of control, the cylinder should not be loaded with more than 50% of the theoretical cylinder force.

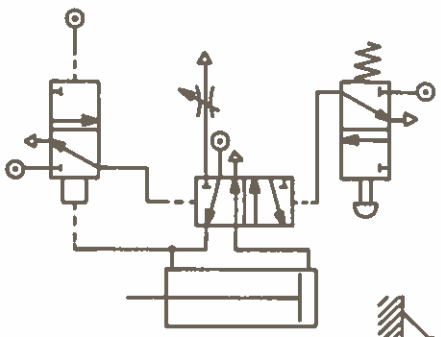


Fig 21

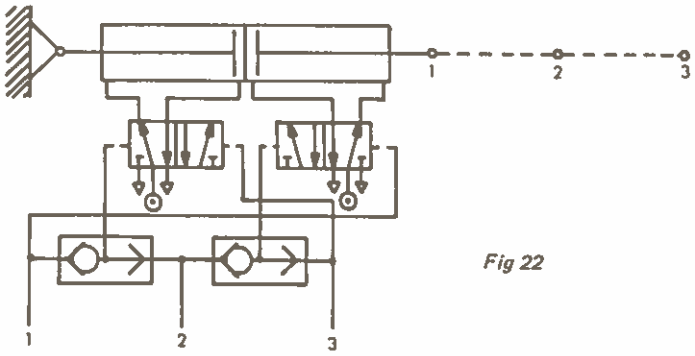


Fig 22

Control of Multi-Position Cylinders

The connection diagram Fig 22 is valid for control of 3-position cylinders and Fig 23 for 4-position cylinders.

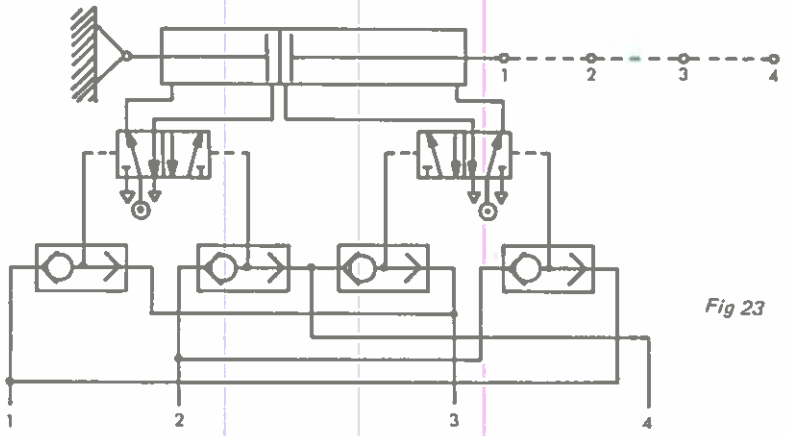


Fig 23

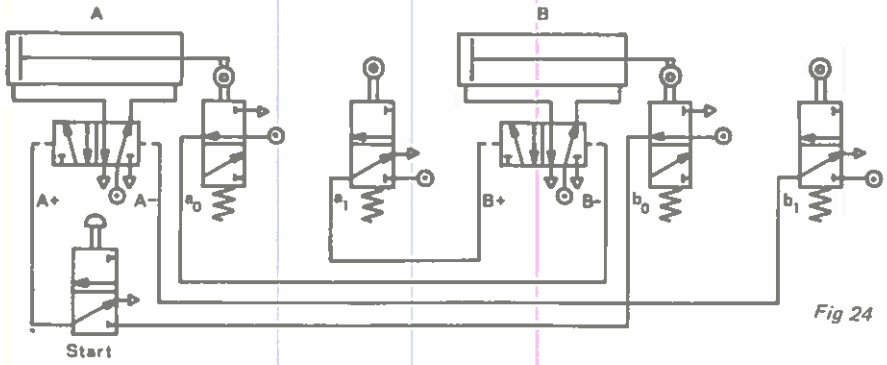


Fig 24

With the connections shown in the figures there is an in-signal for each position the piston can adopt. Irrespective of the position of the piston, a signal to a certain inlet will result in the piston rod adopting the position of the in-signal concerned.

Fig 24 shows a more complex circuit where two cylinders are to be operated in sequence A+, B+, A−, B−. Circuit design becomes increasingly complex when an additional function is required – *eg* to obtain the following sequence A+, B+, B−, A−. Such a circuit is shown in Fig 25, based on the initial premise:

(i) a_0 is connected via 'Start' to A+

(ii) a_1 is connected to B+

(iii) b_1 is connected to B−

(iv) b_0 is connected to A−

To achieve the desired sequence an auxiliary function is to take away signal b_0 from A− before A+ demands a signal. Further, is to take away a_1 from B+ before B− demands a signal. This auxiliary function can be performed by a 5-port 2-position double air-controlled valve as follows:

air is supplied to b_0 while a_1 is bled in one position of the 5-port valve, in the other position, air is supplied to a_1 while b_0 is bled.

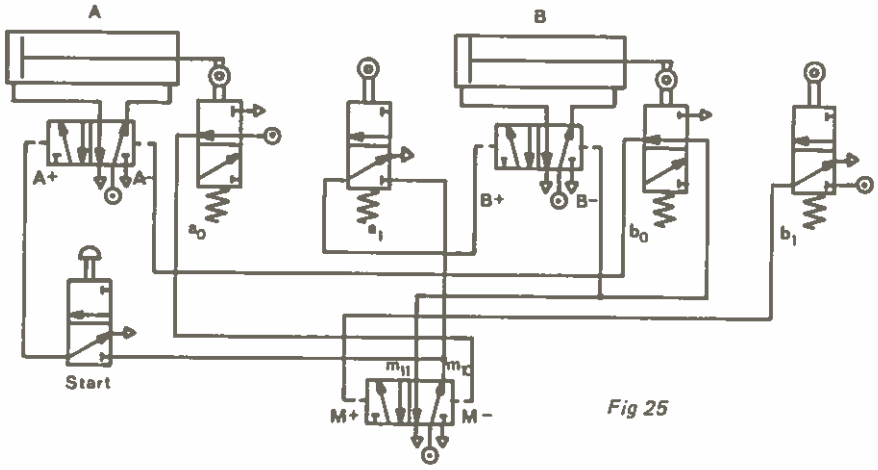


Fig 25

If the connections are made so that when cylinder A becomes minus, a_0 gives a signal to the side of the auxiliary function that bleeds b_0, and when cylinder B becomes plus, b_1 gives a signal to the side that bleeds a_1, the demand is met that b_0 shall be taken away from A– before A+ gets a signal, and that a_1 shall be taken away from B+ before B– gets a signal.

For more complicated sequences it may be necessary to insert further auxiliary valves. A major problem often involved is to appreciate all the blocking signals that have to be broken.

Complex circuit design is normally done on a logic basis – see chapter on *Logic Circuits.*

Air-Hydraulic Circuits

AIR-HYDRAULIC circuits involve the use of air or compressed air or an air cylinder as the direct working unit, with an associated hydraulic circuit providing damping for speed control and uniformity of piston movement. Control is normally obtained via a restrictor valve in the hydraulic circuit. A basic circuit of this type is shown in Fig 1.

The complementary system to this, where hydraulic fluid (oil) is the working medium with compressed air only working indirectly, is known as a hydro-pneumatic circuit. A basic circuit of this type is shown in Fig 2.

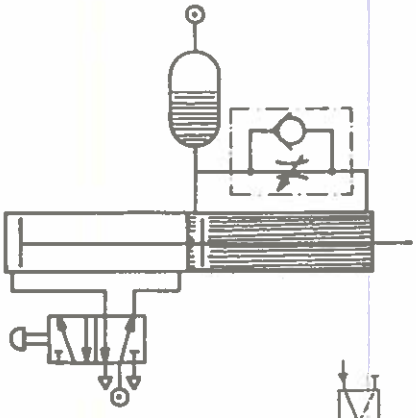


Fig 1

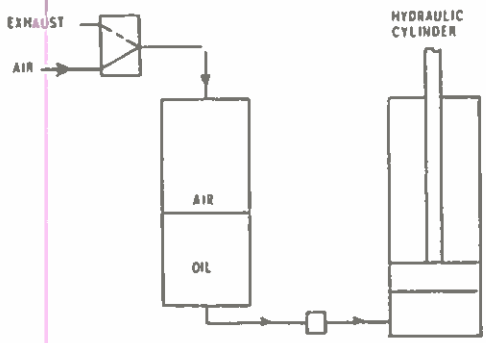


Fig 2

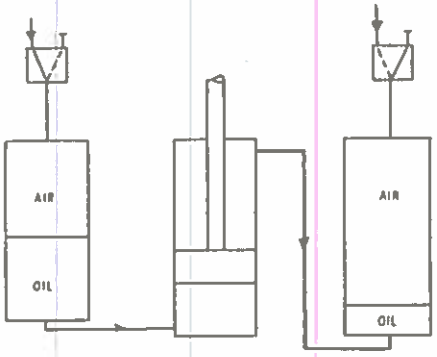


Fig 3

Such a system can be extended to double-acting motion by using a second air/oil reservoir and control valve, as shown in Fig 3. This is further shown in the form of a continuous circuit diagram in Fig 4. Speed control is possible in air in both directions by inserting air-way restrictor valves between the air/oil reservoirs and the hydraulic cylinder.

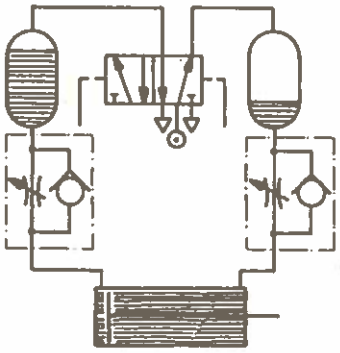


Fig 4

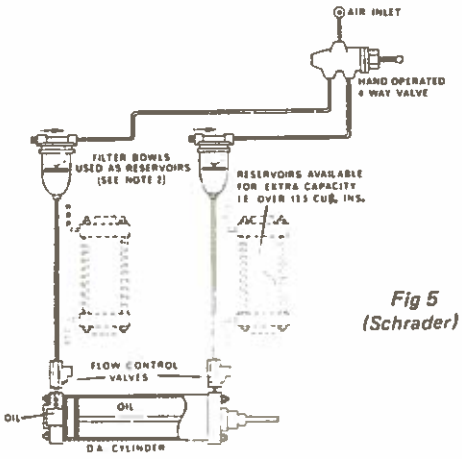

AIR INLET
HAND OPERATED 4 WAY VALVE
FILTER BOWLS USED AS RESERVOIRS (SEE NOTE 2)
RESERVOIRS AVAILABLE FOR EXTRA CAPACITY IF OVER 115 CUB. INS.
FLOW CONTROL VALVES
OIL
OIL
DA CYLINDER

Fig 5 (Schrader)

Air-Hydraulic Conversion

In certain applications requiring rigidity of movement, a hydraulic cylinder can be employed connected indirectly to the compressor air supply thus providing air-hydraulic working without the necessity of using an air-hydraulic cylinder.

The basic arrangement is shown in Fig 5. For small cylinders it may be possible to use the filter bowls as oil reservoirs. If the working volume of the hydraulic cylinder exceeds two thirds of the capacity of the filter bowls, however, it will be necessary to add two oil reservoirs.

The following are important points to note to achieve satisfactory working:-

(i) The hydraulic cylinder should be a low-pressure type with low 'break-out' force and friction. Internal leakage must be minimal, which may call for the use of special seals.

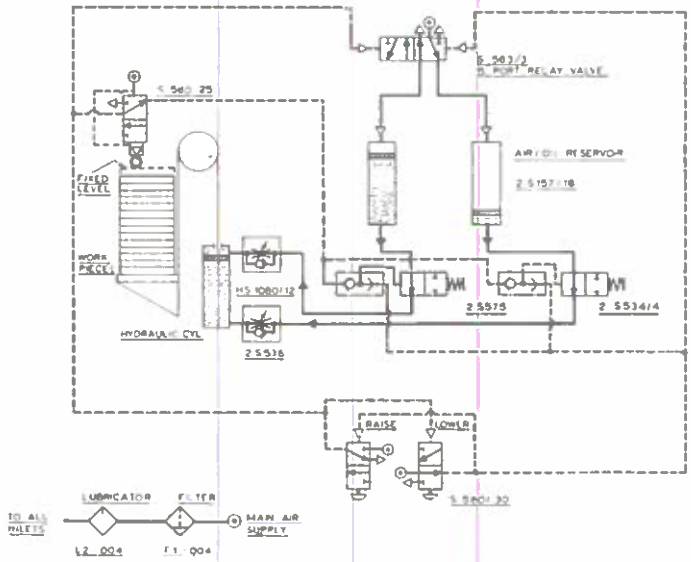
(ii) Filters should be used, even if reservoirs are necessary for additional oil capacity, since these act as baffles to prevent emulsification of the oil.

(iii) If filters only are used, (*ie* if they provide adequate oil capacity), the oil level in either filter bowl should not be allowed to rise above the bottom of the filter element, otherwise oil may be lost through the exhaust port of the main valve.

(iv) Reservoirs should always be mounted at a higher level than the cylinder to provide self-bleeding action when filling.

Air Hydraulic Pumps

The same principle that operates the air-hydro pump can be used to pressurize a hydraulic system, either for general working or for pressure test use. Pressures up to 1000 bar (15000 lb/in^2) can be generated for the latter purpose — very much higher pressures can be generated by specially designed pumps.



Example of a practical double-acting air hydraulic circuit (Martonair).

Small air-hydraulic pumps of this type are widely favoured as direct replacements for normal hydraulic power, both in portable and fixed installations. Air supply can be from compressed air mains, or bottled air (or nitrogen). The fitting of an air pressure regulator enables the hydraulic pressure to be adjusted over a wide range, dependent on the physical size and geometry of the pump.

See also chapter on *Air-Hydraulic Cylinders.*

There's more to air control than the best products in the business.

It can't be said that CompAir Maxam don't go to extremes to be helpful.

They do and every single day of the week too, and this wintry experience shows just how committed Maxam engineers are to getting the job done, come what may.

Somewhere along the line Maxam equipment for a major Scottish production industry in the Highlands had gone adrift.

The emergency plans that followed took the Maxam engineer and the replacement equipment from his Glasgow base to Inverness on the very first flight.

From Inverness he boarded a hire car to find the weather closing in just a mile short of the plant.

That's when the car was abandoned and his long cold trek on foot through the snow started.

On arrival, the precious replacement cylinders were installed in minutes and the plant was on line again.

Little did he know that after trekking back to the hire car he'd have to dig it out!

It's just one of many unusual examples that demonstrate why CompAir Maxam are clearly the leaders in air control.

To meet the more usual demands of production control, Maxam offers industry a diverse and national service.

This includes a well stocked and comprehensive range of pneumatic control equipment.

Most CompAir Maxam products are available for immediate delivery, and you can rely on our delivery times.

The range includes valves, cylinders, logic systems, super speedfit fittings, and a micro-chip programmer for precise low-cost control of very complex production methods.

You'll also find one of our nationwide sales engineers in your region.

We've put him there to give you the very best service and advice you can get hold of.

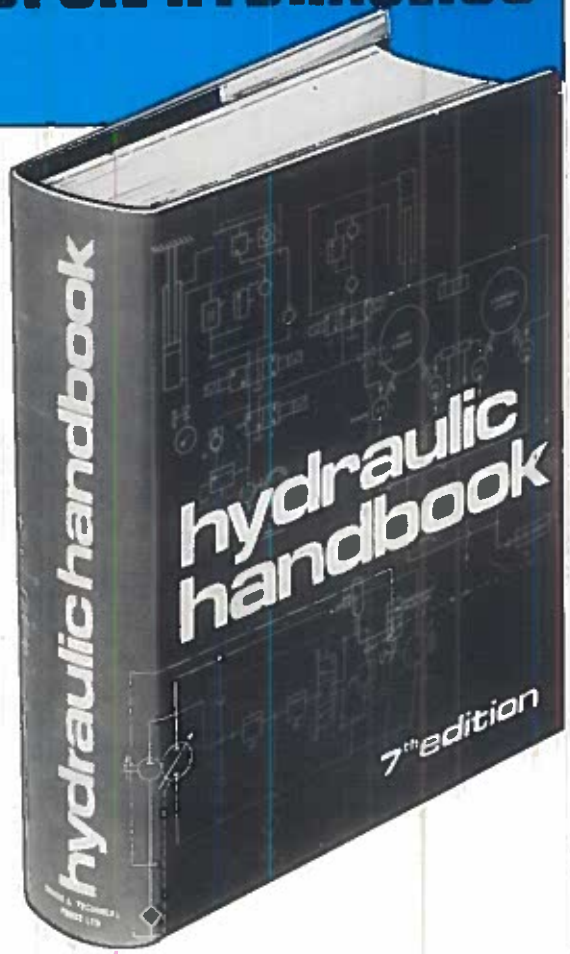
If he can brave the severest of weather conditions to get the job done just think what he can do for you when you put out your cry for help.

For further information call, or write to CompAir Maxam Limited, Pool, Redruth, Cornwall TR15 3PR England. Tel: Camborne (0209) 712712.

CompAir Maxam

Leaders in air control.

THE 'bible' OF OIL-HYDRAULICS



Contents include:-

SECTION 1A – INTRODUCTORY
The Outlook for Hydraulics.

SECTION 1B – PRINCIPLES
Hydrostatics, Hydrodynamics, Properties of Fluids, Actuator Dynamics, Pipework Design and Performance, System Design and Performance, Servo Systems, Fluidics and Fluid Logic.

SECTION 2A – COMPONENTS
Hydraulic Pumps, Hydraulic Power Packs, Pump Selection, Hydraulic Cylinders, Rotary Actuators.

SECTION 2B:
Hydraulic Valves and Selectors, Pressure Vessels, Accumulators, Reservoirs and Tanks, Intensifiers, Pulsation Dampers, Electromodulated Hydraulics.

SECTION 2C:
Pipes and Pipework, Flexible Pipes and Hoses.

SECTION 2D:
Hydraulic Fluids.

SECTION 2E:
Hydraulic Seals.

SECTION 2F:
Plugs.

SECTION 3 – SYSTEMS
Basic Hydraulic Circuits, Controls and Controllers, Fluid Protection, Hydro-pneumatics, High Temperature Hydraulics, Very High Pressure Hydraulics, Miniature Hydraulics, Electrical Specifications.

SECTION 4 – INSTRUMENT TESTING
Instruments, Testing Hydraulic Equipment and Components, Fault Finding and Corrective Maintenance, Vibration in Hydraulic Systems, Servicing and System Maintenance, Noise Control.

SECTION 5 – MACHINES
Hydraulic Motors, Hydrostatic Machines and Transmissions Hydrokinetic Couplings and Transmission, Hydrostatic Bearings.

SECTION 6 – APPLICATIONS
Hydraulics, Automotive, Injection Moulding Machines, Machine Tools, Marine Hydraulics, Mechanical Handling, Mining, Mobile Hydraulics, Workshop Tools.

SECTION 7 – SURVEYS
Survey of Hydraulic Motors, Survey of Hydraulic Pumps, Survey of Hydraulic Cylinders, Survey of Pipe and Hose Fittings, Survey of Hydraulic Valves and Selectors, Survey of Rotary Actuators.

SECTION 8 – TECHNICAL DATA.

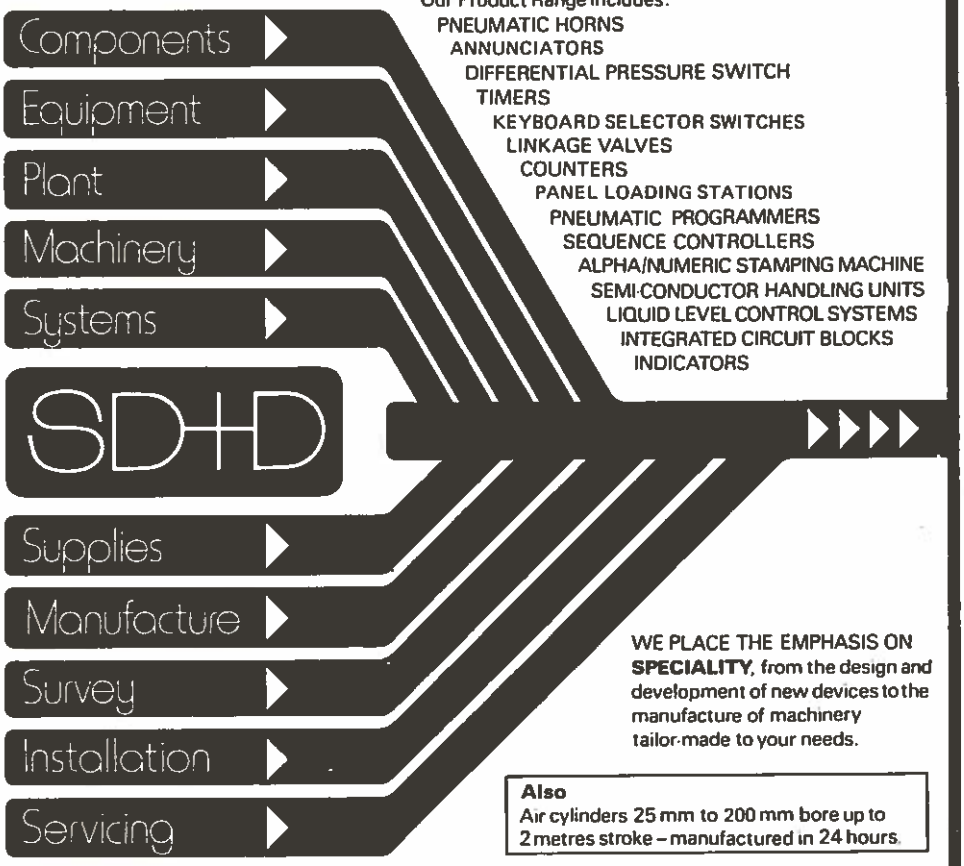
SECTION 9 – BUYERS GUIDE
Trade Names Index, Classified Index of Hydraulic Equipment and Components, Alphabetical List of Manufacturers' Names and Addresses, Index.

850 PAGES
HARD CASE BOUND,
GOLD BLOCKED,
WITH DUST JACKET

TRADE & TECHNICAL PRESS LTD.,CROWN HOUSE,MORDEN,SURREY SM4 5EW,ENGLAND.

Fluid Power Technology

IS OUR SPECIALITY...
applying COMPRESSED AIR-PNEUMATICS-HYDRAULICS
to meet any requirement in any industry.



Our Product Range includes:
PNEUMATIC HORNS
ANNUNCIATORS
DIFFERENTIAL PRESSURE SWITCH
TIMERS
KEYBOARD SELECTOR SWITCHES
LINKAGE VALVES
COUNTERS
PANEL LOADING STATIONS
PNEUMATIC PROGRAMMERS
SEQUENCE CONTROLLERS
ALPHA/NUMERIC STAMPING MACHINE
SEMI-CONDUCTOR HANDLING UNITS
LIQUID LEVEL CONTROL SYSTEMS
INTEGRATED CIRCUIT BLOCKS
INDICATORS

WE PLACE THE EMPHASIS ON **SPECIALITY**, from the design and development of new devices to the manufacture of machinery tailor-made to your needs.

Also
Air cylinders 25 mm to 200 mm bore up to 2 metres stroke – manufactured in 24 hours.

SYSTEMATIC DESIGN & DEVELOPMENTS LTD

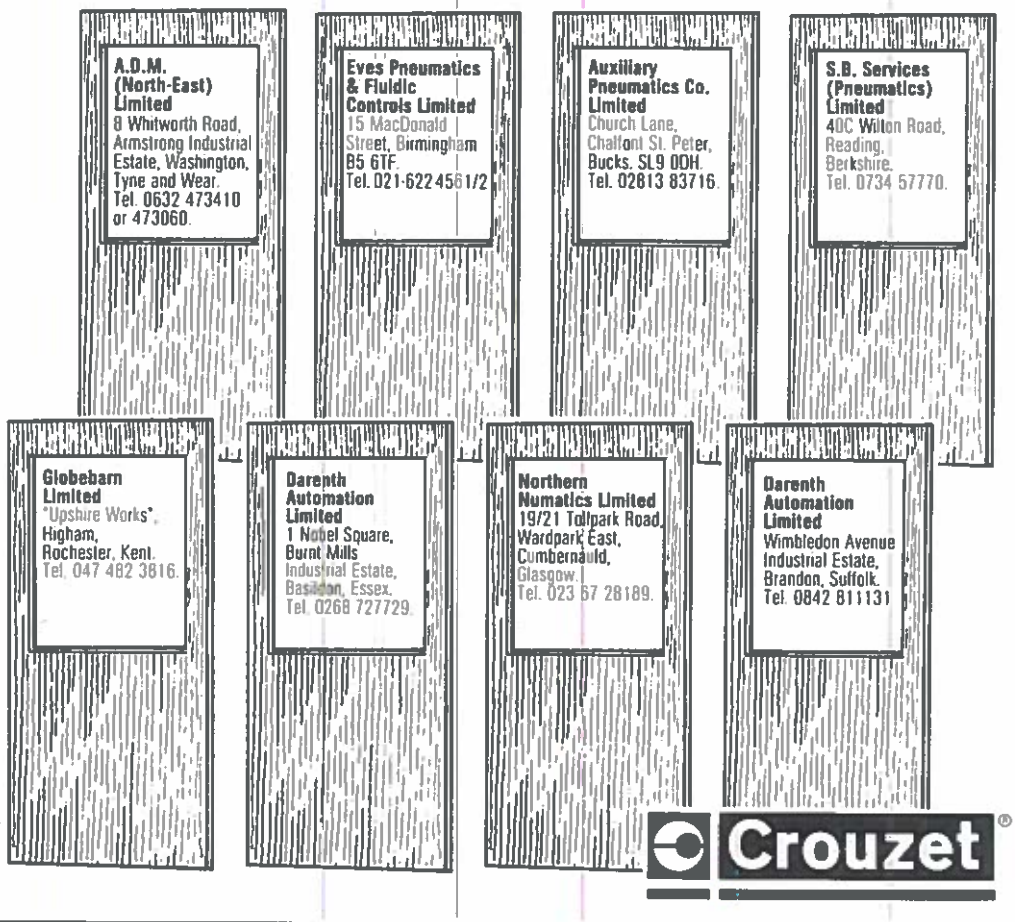
FRIZINGTON ROAD · FRIZINGTON · CUMBRIA CA26 3QY
Telephone (0946) Cleator Moor 810179 (2 lines) Telex 65 294 'SYDED'

Incorporating Pneumethods of Cumbria

Complete pneumatic systems.

Pneumatic logic, control valves, sensors, air-line equipment . . . the lot. Design, production and installation. For your complete pneumatic system and service,

knock on any of these doors, and ask for Crouzet.


A.D.M. (North-East) Limited
8 Whitworth Road, Armstrong Industrial Estate, Washington, Tyne and Wear.
Tel. 0632 473410 or 473060.
Eves Pneumatics & Fluidic Controls Limited
15 MacDonald Street, Birmingham B5 6TF.
Tel. 021-622 4561/2
Auxiliary Pneumatics Co. Limited
Church Lane, Chalfont St. Peter, Bucks. SL9 0DH.
Tel. 02813 83716.
S.B. Services (Pneumatics) Limited
40C Wilton Road, Reading, Berkshire.
Tel. 0734 57770.
Globebarn Limited
"Upshire Works", Higham, Rochester, Kent.
Tel. 047 482 3816.
Darenth Automation Limited
1 Nobel Square, Burnt Mills Industrial Estate, Basildon, Essex.
Tel. 0268 727729.
Northern Numatics Limited
19/21 Tollpark Road, Wardpark East, Cumbernauld, Glasgow.
Tel. 023 67 28189.
Darenth Automation Limited
Wimbledon Avenue Industrial Estate, Brandon, Suffolk.
Tel. 0842 811131

Crouzet®

Crouzet Limited, 108 Hawley Lane, Farnborough, Hants GU14 8HY. Telephone 0252 513211. Telex 858612.

Logic Circuits

LOGICALLY CONSTRUCTED control circuits have numerous advantages over empirically-designed circuits (which are often a combination of intuition and 'guesstimation', finally proven by trial and error). These advantages show up particularly in complex control circuits, where only design by logic can be relied upon to provide the optimal solution.

Logic functions can readily be performed by simple valves or combinations of valves. Interest in pneumatic logic circuits was originally stimulated by the development of pure fluidic devices capable of extreme miniaturization and also incorporating no moving parts. Whilst such a system can be shown to work it has several limitations in practice, notably the fact that it is essentially a low pressure system operating at air pressures of less than 1 bar, consequently needing amplification to power an actuator. Also such a system is critically dependent on air regulation and fine filtering of the air supply. It thus remains mainly of academic interest only.

Its counterpart in providing digital working – or logic circuitry – derived from the miniaturization of more or less conventional valves, *ie* moving part logic (MPL) elements operating at normal system pressure and capable of operating at high speeds. These now form the basis of virtually all practical pneumatic logic circuits, and have been adopted on a relatively large scale. Properly designed LSR elements are reliable, positive in response, and suitable for use with both dry and lubricated compressed air supplies. Also the question of providing interconnections compatible with the size of these elements has been solved by the development of modular construction and assemblies.

The basic problem then remaining is how to design control circuits in terms of logic. The basic 'tools' are Bollean algebra, Karnaugh-Veitch maps and Cascade techniques primarily (there are others). Alternatively there are 'half-way' measures, such as the Step-counter method of electro-mechanical sequencing. All of the basic methods need considerable study to master and apply, and are skills not readily acquired by the intuitive designer.

For that reason, producers of modern MPL components have developed their own system methods to make logic design easier to understand and minimize circuit design time. This simply parallels computer technology where special, and simpler languages have been devised to make programming easier. In that case the language is tied specifically to a particular system, which in turn is based on a particular choice of logic elements providing all the logic functions necessary. This, like the language, can vary with different systems.

Basic Logic Functions

The basic logic functions are NOT, AND and OR, but for logic control functions we also need to add YES (which is only a modified form of NOT). The NOT function can be performed by a 2-way two-position valve, normally open. A signal applied then moves it to closed, *ie* a signal gives no output. There is output only when there is NOT an input.

Simple 2-way valves can be used in combination to provide AND or OR logic response – Fig 1. Equally a 2-way valve can invert an output, to produce the NAND or NOR functions; or invert an input to provide INHIBITION.

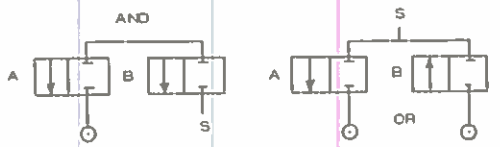


Fig 1

'Logically', in fact, a combination of on–off devices can provide all the logic responses required, except for MEMORY. This would require a combination of 3-way valves. In practical pneumatic logic, however, it is far more realistic to devise single components to provide the logic functions required, which for full coverage of pneumatic logic circuit design requirements are:

NOT AND OR YES NAND NOR INHIBITION and MEMORY

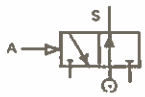
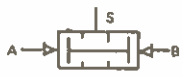
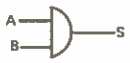
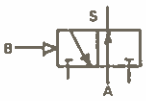
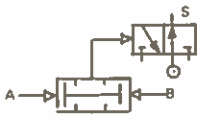
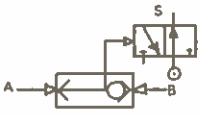
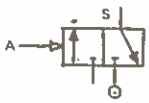
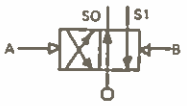
Operating characteristics of these devices are detailed in Table I. Again this listing represents a redundant requirement – *ie* not *all* these functions are necessary to cover all requirements in a logic circuit. The number of individual devices required can be reduced considerably by working with a particular *type* of logic – *eg* AND logic, OR logic, NAND logic or NOR logic; or combinations of these. Certain functions will, however, remain necessarily common with any type of logic chosen – *eg* NOT, INHIBIT and MEMORY. As far as pneumatic logic circuit components are concerned the choice of *type* of logic depends largely on the *suitability* of miniaturized valves to perform the various functions.

If a multiplicity of elements is to be avoided, choice, basically, lies between NOR and NAND logic, or multifunctional devices. NOR and NAND can be made multiple-input devices when, by selecting either one or a number of inputs the other logic functions can be derived. This reduces the number of physical elements required, but each is more expensive in terms of size and cost; and as deployed throughout the circuit, many will be operating with redundant features. Multifunctional devices can provide the basic functions of OR, AND, NOT and MEMORY in smaller (individual) packages; but using 4-way spool valves can present problems in miniaturization.

It is virtually impossible to generalize on this subject, different manufacturers of systems having different methods of approach and component construction. Thus only one system, which has proved particularly acceptable in Europe, will be described in more detail.

'Pneumaid' pneumatic logic units.
(CompAir Maxam)

TABLE I – LOGIC CIRCUIT DEVICES

Logic	Pneumatic Symbol	Logic Symbol	Notes
NOT (3-way valve)			Output is present when there is NOT an input at A NOT A = S or $\overline{A} = S$ $A = \overline{S}$
AND (tandem 2-way valve)			Output is present when there is input at A AND B A *and* B = S or A.B = S
OR (shuttle valve)			Output is present when there is input at A OR B. A *or* B = S or A + B = S
INHIBITION (3-way valve)			Output is present when there is input at B and there is no input at A. In other words, when a signal is applied to A it *inhibits* the signal path of B. B *and not* A = S, or $B.\overline{A} = S$ $B.A = 0$
NAND (tandem 2-way valve and 3-way valve)			Output is present when there is *not* a signal, input at A and *not* a signal input at B. *not* A *and not* B = S, or $\overline{A}.\overline{B}. = S$
NOR (shuttle valve and 3-way valve)			Output is present when there is *not* an input signal at A *or* an input signal at B. *not* A *or not* B = S, or $\overline{A} + \overline{B} = S$
YES (3-way valve)			This is basically an amplifier. Line pressure is connected to B. This pressure is then available at the output when there is a signal input at A. (A = S) In basic terms, this is an inverted NOT with a second input.
MEMORY (4-way valve)			This is a flip-flop device. A signal input at A will give output state S1 and the memory will remain in this state, even when signal A is removed. A signal input at B then changes the output state to S2. It will remain in this state when signal B is removed, until receipt of the next signal A. In other words, the device 'remembers' the last input.

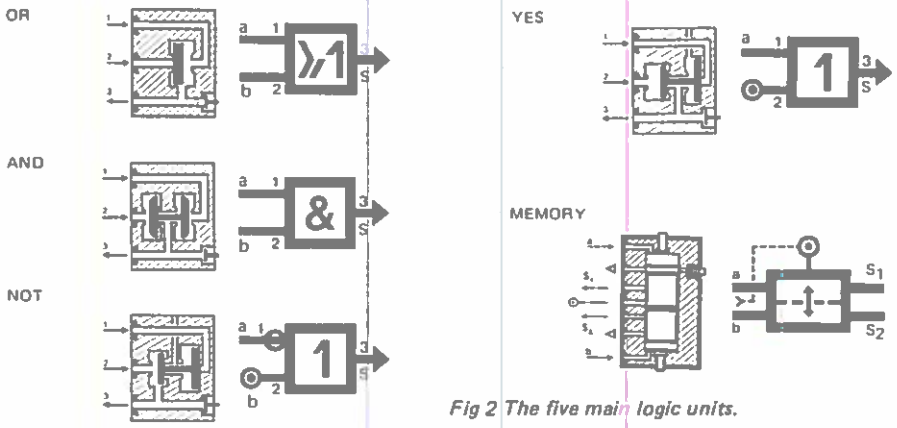


Fig 2 The five main logic units.

Pneumaid System

This system adopts the multi-element approach and is based on five functions — OR, AND, NOT, YES and MEMORY. These devices are shown in Fig 2. It is also allied to a sophisticated mounting system (Polylog) made up of standard modules which can be built up to form a complete monobloc control centre without interconnecting piping.

The devices can be divided into two categories; active and passive. The active devices are basically 3-way, diaphragm-operated, air-return poppet valves, normally open for the NOT function and normally closed for the YES function. These units have separate supply and exhaust ports in addition to the input and output ports.

Passive devices are simple poppet valves actuated by the input signals only, and can perform the AND and OR functions. The MEMORY function is normally supplied by a miniature 4-way slide valve.

All MPL control systems require more than the five basic elements listed earlier. It is possible to produce a number of auxiliary devices from these basic gates.

With a fine restrictor valve having full flow in the reverse direction and a pressure reservoir (or even the capacity of the interconnecting tubing), it is possible to construct simple RC time delays with good repeatability up to 30 s. By combining these elements with the NOT and YES devices both normally open and normally closed delay functions can be achieved.

Similarly if a common signal is fed to the supply and input ports of a normally open delay arrangement, the result is a step-to-pulse converter which can be used to break the trapped signal in the pneumatic system. This is a problem with which most circuit designers are only too familiar. In a number of systems the logic device, reservoir and restrictor are stacked into integral time delay devices which occupy only one position of the mounting matrix.

By the addition of a simple diaphragm/nozzle pilot amplifier stage to the YES unit, a pressure amplifier can be developed which can accept signals from back pressure air sensing units.

The diaphragm-operated NOT unit used in the multi-device logic system has been designed to carry out a function other than that of logic inversion. By setting the pilot pressure to reset at 10% (or less) of the supply pressure, the unit can be used to signal the end of the stroke of a pneumatic cylinder.

There is a refinement of this system in which the NOT unit supply pressure is derived from a port in the cylinder wall set just behind the piston when it has completed its stroke.

Polylog System Design

A circuit using the Polylog system can be designed and built after only a day or two's training. In fact it is possible to design and build the simpler circuit after only two or three hours' tuition. Therefore, no longer is it necessary to be a scholar of Boolean algebra and be able to calculate the resulting equations. Nor does one have to have the knowledge to find one's way round a Karnaugh map.

With the Polylog system there is a direct relationship between the machine cycle, the circuit and the way the elements are assembled. However, to make circuit design easier a simple form of diagram called Grafset is used. This is a European standard to show the graphical representation for expressions of automatic sequences. It is a step/command/transition control graph and a typical diagram is shown in Fig 3. This diagram relates to a circuit involving two cylinders A and B.

The right hand squares represent the movements of the cylinders. A+ refers to cylinder A out-stroking and A– the cylinder instroking *etc*. The squares on the left hand side represent each step in sequence.

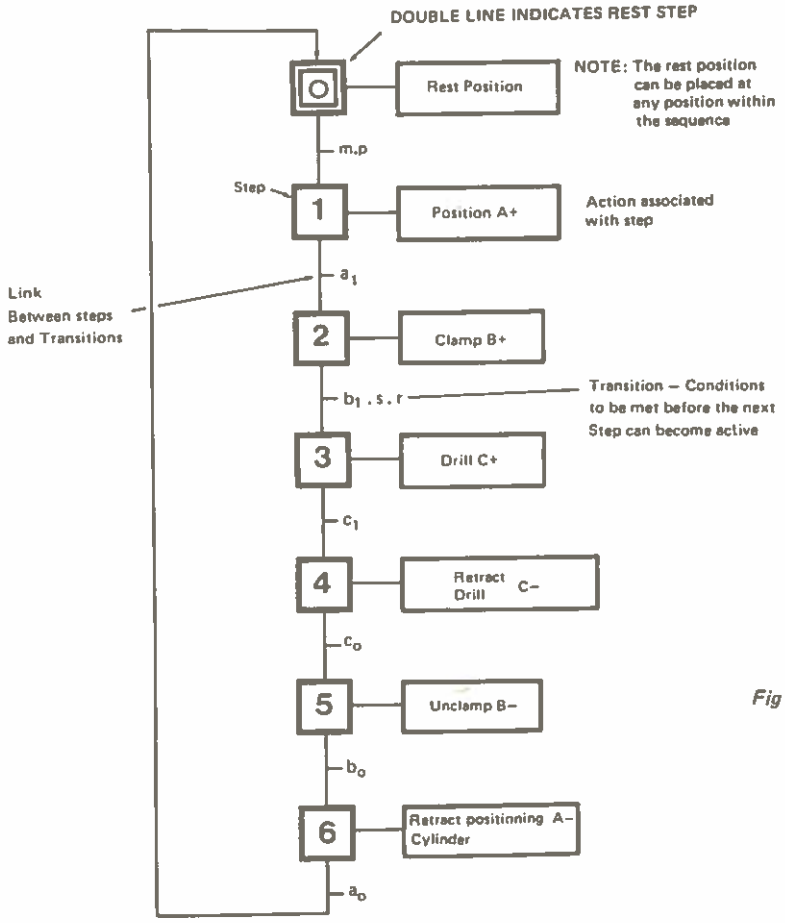
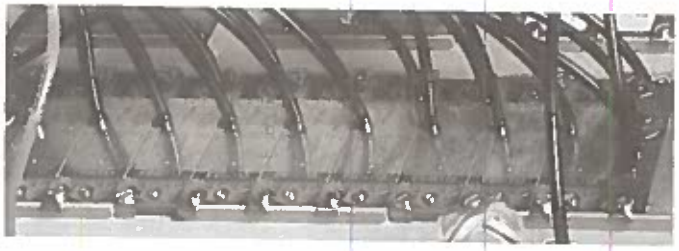


Fig 3



Shift-register chain base mounted. (Festo Pneumatics).

These are the principal steps. Each principal step requires a memory module — thus for this application we would require four memory modules. In between each step there are further letters; these represent interlocks confirming that the action of the principal step is complete or external signals setting the sequence in motion. These are called intermediate steps. This is basically all the circuit design that is necessary before beginning to assemble the system.

In order that the circuit can be quickly confirmed, special cards are available. When placed side by side these represent exactly the layout of the circuit. In this application the cards when put together would be as in Fig 4, shown together with the complete circuit. In this circuit four memory modules provide the four principal steps in the system. This example represents a very simple application.

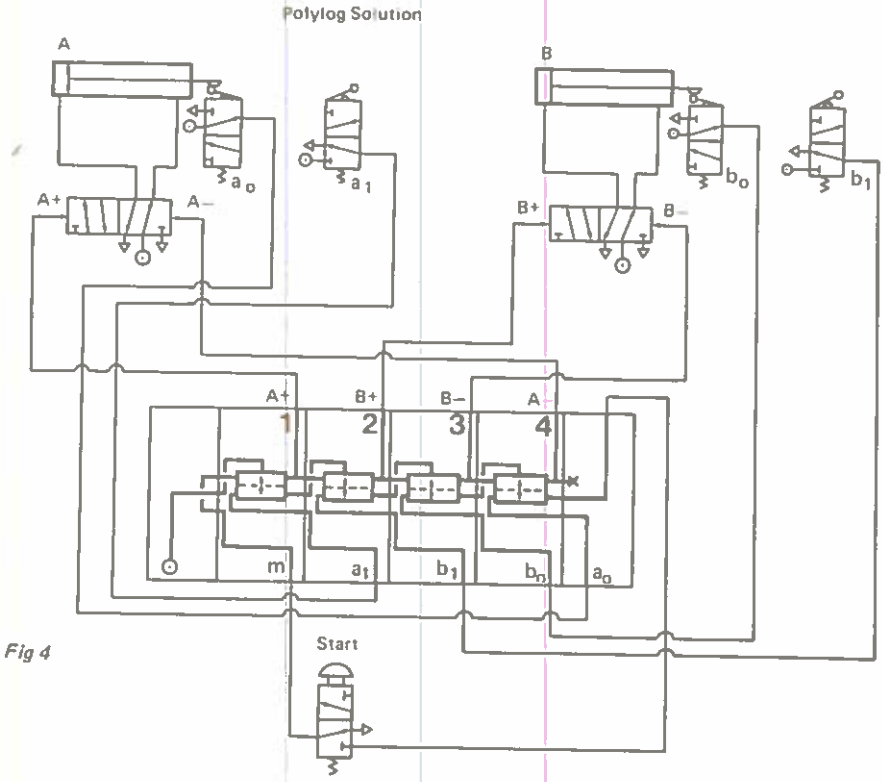


Fig 4

Pneumatic Logic versus Electronics

Pneumatic logic control circuits work on exactly the same principles as programmed electronic controls – both work on a logic basis. Until comparatively recently a pneumatic logic control circuit of medium complexity generally proved cheaper than its electronic equivalent. With the advent of low cost microprocessors the position has changed considerably. Hybrid systems (electronic control of pneumatic systems) have many advantages to offer at comparable, or even lower cost, particularly as the complexity of the control requirement increases. Apart from compactness, the big advantage with a microprocessor is that the programme can be re-written and sequences changed without re-piping the control system.

At the present state of the art, therefore, it would be reasonable to state that where only a relatively few functions are required (say a maximum of 10), pneumatic logic control circuits still have much to offer. Above that number, the Programmable Logic Controller (microprocessor) offers increasing overall advantages. Where pneumatic controllers can continue to remain unchallenged is in high-risk fire areas, or hazardous areas – *eg* involving low and high flashpoint materials in processing.

System Maintenance

MAINTENANCE REQUIREMENTS fall under three main headings, in the following order:

(i) Maintenance of tools and appliances.

(ii) System maintenance.

(iii) Compressor maintenance.

All are equally important, but require different maintenance intervals. The life of pneumatic tools and appliances is generally appreciably lower than that of a compressor, hence the need for more regular preventative routine maintenance.

Air Tool Maintenance (see also Table I)

A programme for air tool maintenance should cover the following points:

(i) Ensuring that the air supply is maintained at the required pressure and with the required delivery.

(ii) Regular checking and refilling of lubricators in order to maintain adequate lubrication.

(iii) Checking individual tools for handling or manipulative damage.

(iv) Individual tool servicing as required.

(v) Checking that tools are handled in a reasonable manner (*eg* cleaned after use) and used in the proper manner (*eg* not abused).

The most common cause of early troubles with workshop tools is lack of lubrication. The next is abuse, such as tools being dropped or dragged around by the hose, or generally ill-used. The design of the work area itself, and particularly the allocation of racks for holding tools when not in use, can in itself be a form of preventative maintenance.

Scheduled maintenance should, preferably, be the responsibility of a single individual rather than part of the tool operator's job. With adequate records and experience, tools can then be withdrawn at specified intervals for stripping and inspection and the replacement of worn or damaged parts, to maintain a high operating efficiency. This applies generally to all forms of pneumatic tool. Records may also provide a valuable pointer as to where (which operator) or under what conditions individual tools are suffering abuse, as shown by higher wear or damage rates, and remedial action should be taken as appropriate. This may be an operator fault or a tool fault (*eg* the tool not adequate for the job at a particular station).

TABLE I – AIR TOOL MAINTENANCE POINTS

Check or Fault	Likely Cause	Action
Dirt or gummy deposits on motor	Insufficient or wrong oil; dirty air	Check lubricator. Check line filter.
High blade wear on motor	Insufficient oil; normal wear	Reduce periods between scheduled maintenance.
Bearing play	Excessive load; normal wear	Check tool capacity for job; replace bearing
Bearing rough	Dirty air Lack of lubrication Normal wear	Check filter Check lubricator Replace bearing
Governor action sluggish	Air leakage	Check governor bushings for wear
Loss of power	Low air pressure Air leakage Governor fault	Check line pressure and filter Check and rectify Check governor condition; adjust action if necessary
Vibration	Breakage	Replace faulty part(s)
Loss of speed	Low air pressure Air leakage Governor fault	Check line pressure and filter Check and rectify Check

In the case of major repairs, tools would normally be returned to the manufacturer. With adequate preventative maintenance and efficient scheduled maintenance, a major breakdown is unlikely to occur until the tool is nearing the end of its useful life. The true value of the tool is represented by its initial cost less the amortization rate and if the major repair cost exceeds the value of the tool at that time it would be better replaced by a new one. This applies particularly in the case of workshop tools.

Appliance Maintenance

Maintenance overhaul of air-operated appliances requires the following precautions:

(i) All air pressure should be completely released from the system.
(ii) It should be impossible to start the appliance inadvertently.
(iii) The correct tools should be available for the job.
(iv) Adequate lifting gear should be available (if necessary) and checked to be in good condition.

After maintenance work, and before clearing the machine for use, check:

(i) That operating pressures, speed and working temperatures are correct.
(ii) That controls and shut-down devices work correctly.
(iii) That the sump is filled with the correct grade of oil (where appropriate).
(iv) That there is full flow of coolant (when appropriate).
(v) Condition of discharge pipe and pulsation chamber (if fitted); clean off any excess deposits.

TABLE II – TROUBLE-SHOOTING – AIR CYLINDERS

Fault	Cause	Action
Loss of thrust*	(i) leaking piston seals	Replace seals
	(ii) corrosion on bore	Refinish bore as necessary to eliminate wear on seals
	(iii) dirt trapped in seals	Replace seals. Clean cylinder. Check condition of rod wiper seal. Fit external gaiter if necessary for better protection.
	(iv) excessive friction	Check condition of hose, seals, wear rings and rod bearing
	(v) low pressure	Check supply pressure at inlet
Leakage past rod	Rod seals faulty	Replace seals unless compression type seals are fitted in an adjustable gland. In that case, tighten gland.
Loss of cushioning	(i) cushion valve blocked	Remove and clean
	(ii) cushion seals failed	Replace cushion seal(s)

*Loss of thrust can also result from substantially increasing the speed of working with no fault present in the cylinder or system.

Air Cylinder Maintenance (see also Table II)

Air cylinder maintenance requirements should, in general, be minimal, except for the periodic replacement of expendable parts. These are normally confined to piston seals, end cover seals (where applicable), rod seals and rod bearings. The latter may be in the form of a replaceable cartridge assembly.

The period required between servicing will depend on the frequency with which the cylinder is used and the total number of operations, and the working conditions. Scraper rings and, if necessary, an external gaiter, together with the provision of adequate lubrication against corrosion, can protect the inside of the cylinder against abrasive atmospheric contaminants, provided the lubricant used is resistant to water-washing. Most modern cylinders, too, are precoated during assembly with a special lubricating compound which normally adheres to the surfaces for a considerable period. Thus, even in the absence of air line lubrication, good internal working conditions are established between servicing periods with a minimum of wear. Separate, continuous lubrication is, however, always recommended for the highest degree of reliability and minimum wear. This can considerably extend the period between servicing.

Seal wear, and thus seal life and performance, is very largely dependent on the bore condition. Corrosion, pitting, abrasion or damage to the bore can result in high seal wear and premature failure. Intermittent use, with long periods of idleness during which seals may dry out completely, is also damaging. Most seal failure, however, can usually be traced to bad initial fitting of the seal, the use of the wrong type of seal, or wrong material specification.

Piston seal leak is not always easy to detect, since the first evidence may appear as leakage at the exhaust ports or a control valve, when a valve seal rather than the piston seal may at first be suspect.

A simple check is to disconnect the exhaust line from the cylinder to the valve. If there is leakage of air through this line, the cylinder piston seals are faulty. If there is no leakage from this line, the presence of leakage at the exhaust port of the valve indicates internal valve leakage.

Tie-rod construction generally makes cylinder disassembly easy. A point to watch with single-acting spring-return cylinders is that one end cover will be under spring load and needs backing off carefully until the compression of the spring is relieved. On longer stroke spring-return cylinders it may be advantageous to extend the length of the tie rods to accommodate the full (uncompressed) spring length with the cover still on the tie rods. Nuts can also be welded onto the opposite ends of the tie rods so that disassembly can only be attempted from the extended tie rod end.

Replacement of piston seals may or may not require disassembly of the piston, depending on the type of seal fitted and the manner in which they are retained. Every precaution should be observed in refitting seals to avoid stretching, nicking or otherwise damaging them. End cover seals (where applicable) should always be replaced with new ones when a cylinder is dismantled and reassembled. A general lubrication of the cylinder before assembly is also recommended.

System Maintenance (see also Table III)

Scheduled periods for the inspection of drains, moisture traps and separators should be based on experience with the particular components involved (manufacturer's recommendations should be used as an initial guide).

Filters and filter/separators fitted at individual tool stations must receive regular attention, particularly if they are of the manual drain type. The level of contaminants will normally be visible in the filter bowl or an excessively high level indicated by a tell-tale. Both types — manual drain or automatic — will also require periodic cleaning of the filter element; thus a regular period must be set down, depending largely on the 'dirtiness' of the system, for cleaning or replacement of the filter elements as appropriate. Some types of filter have an element which can be cleaned very simply by blowing out; others have an element which can only be cleaned by special techniques (and which thus need replacement whilst the element is returned to the manufacturers for cleaning); still others have elements which are discarded and replaced after a specified period of use. A filter element accidentally damaged during cleaning should always be replaced by a new element.

Clogged filters are the most common source of pressure drop at tool points. Air leaks or constrictions in the flexible line are other obvious sources of pressure drop at tool points. The former can normally be heard with the tool connected but switched off. Kinks or damage to hose that has not produced puncturing may show up in the form of variable performance as the tool position is changed. In this case end fittings should first be checked for air tightness, then the hose length itself checked for damage.

Less obvious sources of pressure loss may occur farther back in the system. With adequate line sizing and everything in good condition, pressure drop between the air receiver and the tool station take-off point should not normally exceed 0.2—0.35 bar (3—5 lb/in^2) (or in some installations 10% of the nominal system pressure). Additional losses at the tool point itself should not normally exceed 0.35 bar (5 lb/in^2), although again a maximum figure of 15% total pressure drop to the tool is sometimes accepted.

Regardless of the actual system pressure drop as installed and originally worked, any further loss of pressure at take-off points or at individual tools is a subject for immediate attention so that the fault can be rectified. Loss of pressure not only reduces tool working efficiency, but may also be expensive in terms of compressed air wastage if due to a leak. Leakage may develop at joints, connections, *etc*, or at traps where the valve has been jammed open by contaminants, apart from such obvious sources as mechanical damage to the lines.

TABLE III – MAINTENANCE POINTS – PIPEWORK

Type of Line	Possible Cause of Trouble	Action
Rigid line system	(i) dead weight of piping	Support with pipe clips, closer spaced on horizontal runs
	(ii) expansion and contraction	(a) use clips which allow lateral movement of pipes (b) incorporate bends to accommodate movement where there is a large difference in ambient temperatures
	(iii) internal pressure	Provide adequate support as necessary to prevent movement and flexing
	(iv) leakage	(a) all pipe joints to be properly made (b) replace faulty valves or fittings (c) if caused by damage check environmental conditions; re-site or protect vulnerable fittings
	(v) water entrainment	Check that suitable water traps are provided and that their location is satisfactory. Establish suitable draining period and check period for manual traps. All loop bends should be upward facing, not downward.
Flexible lines	(i) leakage	(a) check for wear or deterioration at end couplings (b) re-site or protect hoses subjected to bad environmental conditions (c) lay down standard procedure for manipulation of flexible hoses (d) consider possible use of automatic close coil flexible lines
	(ii) excessive pressure drop	(a) check hose for bore condition (b) check that suitable size of hose is being used

It is not commonly appreciated that an excessive line pressure can be as bad as lack of line pressure. Whilst the latter reduces the working efficiency of the tool – a tool designed to operate on 5.6–7 bar (80–100 lb/in^2), for example, may suffer a loss of efficiency of 25% operating on 5 bar (70 lb/in^2) – excessive pressure results in accelerated wear and possible over-stressing of parts, which could lead to early failure.

Lubricators at tool points also need regular attention to ensure that an adequate supply of oil is maintained – and the rate of oil injection can also be checked if necessary. It is particularly important that only the specified oil be used to refill the lubricators as an oil of different viscosity may not only upset the lubrication rate but lack the qualities necessary for a pneumatic tool lubricant. Light oils of approximately SAE 10 viscosity are normally specified, specially compounded for pneumatic system use (*eg* normally containing emulsifying agents and rust inhibitors).

Pipework should require only minimum attention — *eg* inspection for possible leaks. The following may, however, show up as *installation faults* needing correction.

(i) Sagging or displaced pipes — requiring proper support with pipe clips, *etc.*

(ii) Evidence of whipping at bends — again due to inadequate pipe support under pressure surges.

(iii) Distortion of runs — evidence of expansion/contraction movement which should be accommodated by bends or expansion joints.

Compressor Maintenance (see also Tables IVA, IVB, IVC and V).

Compressor maintenance is usually specific to the type of compressor and the installation. Basic requirements in all cases are cleanliness (*eg* regular cleaning of intake filters), regular attention to drain points as appropriate (*eg* on the compressor itself if any, intercoolers and receiver) and maintenance of an adequate supply of lubricant. Other basic requirements involve maintaining the correct tensioning of intercooler and main driving belts, and routine oil changes. Some compressors may also be fitted with intake silencers which may need either periodic cleaning or replacement of felt layers at regular intervals in order to maintain volumetric efficiency. The valves of a reciprocating compressor should be inspected and cleaned in accordance with the maker's instructions and as frequently as the duty necessitates.

TABLE IVA — MAINTENANCE POINTS — RECIPROCATING COMPRESSORS

Component	Interval*	Action
Air filter (inlet)	Fortnightly	Clean as necessary
Cooling system	Continuous	Check water temperatures
	As necessary	Clean water side on water-cooled systems
Lubrication	Periodically*	(i) Check cylinder oil feed rate (ii) Check oil levels (iii) Change crankcase oil
Bearings	Periodically*	Check wear. Adjust for wear or replace as necessary
Drains	Hourly*	Intercooler drains
	Daily*	Other drains or all automatic drains
Valves	Periodically*	Check for condition and replace plates or springs as necessary
Safety valves	Periodically*	Clean and inspect
Piston rings	Annually	Inspect for condition and replace as necessary
Packing glands	Periodically*	Adjust if applicable; otherwise inspect for condition and replace if necessary

*Suitable intervals can only be established by experience or on manufacturer's recommendations

TABLE IVB – OIL CHANGES – RECIPROCATING COMPRESSORS

Compressor Type	Working Conditions	Change Interval
All stationary	Running in	After 100 hours
All portable	Running in	After 50 hours
Stationary	Clean atmosphere	6 months or 2 000 hours
	Dirty atmosphere	3 months or 1 000 hours*
Portable	Average	1 month or 500 hours
	Dirty atmosphere	2 weeks or 250 hours*
	Very dirty atmosphere	1 week or 100 hours*

*May be adjusted by experience

TABLE IVC – TROUBLE-SHOOTING – RECIPROCATING COMPRESSORS

Fault or Symptom	Possible Causes	Action
'Knocking' or heavy vibration	(i) drive or coupling elements loose	Check and rectify
	(ii) unstable oil-relief valve	Causing hydraulic hammer
	(iii) excessive working clearances on crosshead or bearings	(a) Check for wear (b) Check for mis-alignment
	(iv) cylinder clearance volume too small	Check for accumulation of water, oil or other deposits
Abnormal air temperature		See OVERHEATING (check at inter-stages as well as at delivery)
Abnormal air pressure	(i) leakage	Find and correct source of leakage
	(ii) loss of compression efficiency	See EXCESSIVE CYLINDER WEAR
	(iii) seal failure	Check compression seals
Whistling noise	(i) valve leakage	Identified as a high pitched whistle varying in intensity with pressure
	(ii) gland leakage	
	(iii) leakage in connections	Identified as a hiss
Rumbling noise	(i) air leakage past piston rings	
	(ii) lack of lubrication	
Valve chatter	Defective or broken valve springs	Replace springs

TABLE V – MAINTENANCE POINTS – ROTARY COMPRESSORS

Component	Inspection	Further Action
Electric motor	Clean motor; inspect for wear; check bearings; check clearances; check general condition	As necessary
Couplings (where appropriate)	Clean and examine	Replace if necessary
Casing	Open and examine for corrosion and wear	
Journal bearings	Examine for wear	Replace, reline, adjust with shims } as necessary
Shaft seals	Check condition of clearances	Replace if necessary
Rotor	Examine for corrosion, pitting and wear; check clearances; check balance	As necessary
Gear case (where appropriate)	Examine for corrosion and wear; check damper	As necessary
Other parts	Examine for corrosion, erosion and wear	As necessary
Governor/Regulator	Clean and inspect; check for wear; check adjustment	Replace worn parts; adjust for proper operation
Auxiliary drives	Check	Replace worn parts
Instrumentation	Check for operation and correct reading	Recalibrate or adjust as necessary

Compressor manufacturers normally specify routine maintenance requirements in detail in a manual appropriate to the machine. They may also list recommended spares to be carried, with an indication of which may be regarded as replaceable components. Servicing and attention required will also vary widely with different types of machine. Thus rotating machines normally need more detailed maintenance and inspection than reciprocating machines.

SECTION 4a

Tools

A new generation of pneumatic screwdrivers

FIAM have developed a new range of pneumatic screw driving machines with very interesting features.

Energy saving: the automatic clutch disengaging device ensures an outstanding reduction in compressed air consumption and less component wear.

Operator benefits: minimum noise when machine is running, absence of vibration and shock, reduced atmospheric pollution, very good handling characteristics.

Performance: accurate control of the tightening torque with no torque fluctuation; simple adjustment over a wide speed range; easy adjustment of the direction of rotation, of the closing torque, of the starting system and of the handgrip; wide range of applications: screws, nuts and bolts up to M6, even with brittle and soft materials.

Productivity: reduction of discards and rejects thanks to special features such as immediate torque increase, reverse in direct drive for unscrewing in special circumstances; reduction of assembly time thanks to automatic propulsion starting, to easy operation and to automatic stopping when screwing ceases.

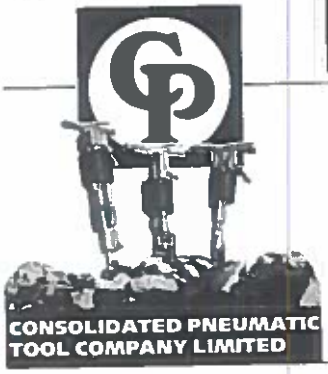
Do not hesitate to consult our head office; a new generation of screw driving machines only needs to be tested to be believed.

pneumatic tools
FIAM SPA
Viale Crispi 123 - 36100 VICENZA/ITALY
Tel. (0444) 562611 - Telex 480195

but whatever you do...

CP it.

Mining and construction air tools and compressors from Consolidated Pneumatic. The right tool for every job, and the right place for every tool.



Consolidated Pneumatic Tool Co. Ltd. (Trading Division)
Greenwell Road,
Aberdeen AB9 2JQ
Telephone (0224) 871901
Telex 73370

MAIN DEALERS

Glasgow
Stewart Plant Sales Ltd.
Edinburgh
Stewart Plant Sales Ltd.
Dundee
Stewart Plant Sales Ltd.
Aberdeen
Stewart Plant Sales Ltd.
Ross-shire
Stewart Plant Sales Ltd.
Manchester
Scot Plant Sales Ltd.
Leeds
Scot Plant Sales Ltd.
Tyne and Wear
Scot Plant Sales Ltd.
Midlands
Gunn Plant Sales Ltd.
Staffs.
Gunn Plant Sales Ltd.
South West
W. R. Selwood
South Wales
Davies Plant Sales
Belfast
Blackwood Hodge (Ireland) Ltd.
Dublin
Blackwood Hodge (Ireland) Ltd.

OFFICIAL DISTRIBUTORS

London & South East
Oswald Record Ltd.
Marmax Engineering Co. Ltd.
Rawley Plant Ltd.
Builders Equipment Ltd

Pneumatic Tools

ALL TOOLS powered by compressed air have the advantage of being stall-proof, *ie* they will not be damaged if overloaded. This is a particularly useful characteristic with motor driven tools.

Pneumatic tools can be classified by basic mechanical type, *eg* percussive, rotary, or air motor driven — which largely governs their working characteristics; or more usefully by intended duty. The latter fall into three broad groups — see Table I. Of these contractors' tools and mining tools have much in common. Workshop tools embrace a much wider range of purpose-designed tools for manual, semi-automated and fully automated use.

Percussive Tools

The theoretical energy (e) per blow delivered by a percussive tool is:

$$e = \frac{m \cdot v^2}{2} \text{ Nm}$$

where m is the mass of the piston
v is the velocity with which the piston strikes the tool shank

The theoretical power delivered, at an impact rate of n/s, is thus:

$$\text{power} = \frac{m \cdot v^2 \cdot n}{2s} \text{ Nm}$$

In practice, energy per blow is usually the most important parameter, since this needs to be selected to suit the work requirements, but the impact rate may also be significant because of its effect on material characteristics (*eg* in riveting). The stroke of the piston may also be a significant factor in the design of the tool in order to obtain the required output. The impact energy achieved may also be limited by the amount of recoil which can be tolerated without making the tool difficult to handle.

In order to produce oscillation of a piston in a percussion type of tool, suitable porting or valving is necessary to deliver air under pressure to alternate sides of the piston. In the case of light tools, and where a high rate of oscillation is desirable (as in scaling hammers and dressing hammers), compressed air intake and exhaust may be achieved via ports in the cylinder wall, the openings of which are controlled by movements of the piston. Such types are sometimes referred

TABLE I – CLASSIFICATION OF AIR TOOLS BY PURPOSE

Type	SUB-GROUPING		
	Contractors' Tools	Mining Tools	Workshop Tools
Percussive	Breakers Diggers Tampers Hammers Spaders Shuttering vibrators Stone working tools Road rippers Pile drivers Picks Steel extractors	Rock drills Coal picks Drifters Stopers Breakers Diggers Spaders Sheeting drivers Spike drivers Steel extractors	Scaling hammers Caulking hammers Chipping hammers Riveting hammers Weld flux scalers Paint scrapers Chisels Vibrators Engraving tools Rivet cutters Rivet busters
Rotary	Shank grinders Concrete vibrators Sump pumps Sludge pumps Conveyors Rock drills	Coal drills Stone drills Vibrators Sludge pumps Sump pumps Air motors	Grinders Pencil grinders Vibrators Screwdrivers Woodborers Deburrers Shears Nibblers Tappers Reamers Nut runners Stud runners Rotary files Burrs Impact wrenches Tube expanders
Machines	Compressors Portable compressors	Compressors Portable compressors	Compressors Portable compressors
Air motor driven	Concrete vibrators Hoists Winches Pumps	Vibrators Riddles Conveyors Loaders Coal tables Pumps Hoists	Vibrators Sifters Hammers Hoists Winches Pumps Tube expanders
Cylinders	Tool feeds Jacks Hoists Air legs	Tool feeds Jacks Hoists	Clamps Vices Forging hammers Fettling grips Chucks Jacks

to as valveless tools. For heavier tools or where better control is required, mechanical valving is normally employed, of which there is a considerable variety in design. In general, however, the three main types are:

(i) Thimble or tubular valves.

(ii) Plate valves.

(iii) Flapper valves

Tubular valves are extremely straightforward in construction and work on the 'compression throw' principle with a high flow efficiency. Their chief limitation is that their relatively high mass limits the speed at which satisfactory valve action can be maintained, hence they are less suited for fast hitting hammers. Plate type (circular disc) and flat type valves are generally quicker acting, but tend to have a somewhat lower flow efficiency. The type and design of the valve chosen is, therefore, usually representative of the manufacturer's own solution to the necessary compromise between flow efficiency and operating speed, consistent with the requirements of the particular tool involved.

The majority of roadbreaking, contractors' and mining tools powered by compressed air are of the percussive or reciprocating type. They work on the principle of impingement of compressed air on alternate sides of a piston confined within a cylinder, the reciprocating or oscillating motion of the piston being used to actuate the tool. The tool itself may either be an extension of the piston or a rigid attachment to the piston, as in the case of hammer-type tools; or the work head may form a separate part of the tool detached from the piston and free to move independently of the head, as in the case of rock drills, riveters and the bulk of the heavier percussive tools. The majority of modern rock drills, *etc*, also incorporate a self-rotating action, and thus differ in this respect from concrete breakers.

Consolidated Pneumatic portable screw compressor in use with a CP breaker.

TABLE II – TYPICAL MAXIMUM AIR CONSUMPTION OF PNEUMATIC TOOLS

	m^3 free air per minute at 6 bar (gauge)	Cubic feet free air per minute on full throttle 80 lb/in^2 (gauge)
Rock Drills		
Drifter drill (cradle mounted), 3 in	4–4.8	140–170
Drifter drill (cradle mounted), 3½ in	4.2–6	150–210
Drifter drill (cradle mounted), 4 in	5.7–8	200–280
Wagon drill with 4 in drifter	6	210
Wagon drill with 3½ in drifter	5	175
Hand hammer drill (jack hammer), 28 lb	1.5	50
Hand hammer drill (jack hammer), 35 lb	1.7	60
Hand hammer drill (jack hammer), 45 lb	2.25	80
Hand hammer drill (jack hammer), 60 lb	4.25	150
Plug drill	0.85	30
Sinker drill, 3 in	3.5	125
Sinker drill, 3½ in	4.5	160
Stoping drill, light	2	70
Stoping drill, heavy	3.5	125
Dustless dry drill, 3½ in	5.1	180
Dustless dry drill, 3 in	3.4	120
Underwater drill	2.4–2.6	85–90
Pneumatic Tools (Percussive)		
Concrete breaker, 75–80 lb	1.8–2	65–75
Concrete breaker, 50 lb	1.4	50
Concrete breaker, 30 lb	1.2	40
Pile driver	1.7	60
Chipping and caulking hammer, light	0.3–0.35	10–13
Chipping and caulking hammer, medium	0.5	18
Chipping and caulking hammer, heavy (suitable also for ½ in hot rivets)	0.6	23
Riveting hammer, ¾ in, hot rivets	0.7	24
Riveting hammer, 1 in hot rivets	0.8	26
Riveting hammer, 1¼ in hot rivets	0.9	29
Riveting hammer, 1¾ in hot rivets	1.1	35
Rivet buster, heavy single blow type	0.09 per rivet	3 per rivet
Rivet cutter, rapid blow type for rivets up to ¾ in	1.1	35
Single blow riveter (one shot for aircraft rivets)	0.002 per rivet	1/16 per rivet
Staybolt riveter for 1 in copper staybolts	0.9	33
Stand riveter for ½ in hot rivets	0.7	24
Stand riveter for ¾ in hot rivets	0.8	26
Stand riveter for 1 in hot rivets	0.85	28
Jam riveters as for stand riveters		
Yoke riveters as for stand riveters		
Hammer holder-on	0.85	28
Holder-on	0.009 per rivet	1/8 per rivet
Wood deck caulker	0.6	22

cont.....

TABLE II – TYPICAL MAXIMUM AIR CONSUMPTION OF PNEUMATIC TOOLS (contd).....

	m^3 free air per minute at 6 bar (gauge)	Cubic feet free air per minute on full throttle 80 lb/in^2 (gauge)
Pneumatic Tools (Percussive) ...continued		
Pick, light	0.6–0.85	20–30
Pick, medium	0.85–1.2	30–40
Pick, heavy	0.7–1.1	25–35
Spader, light	0.6–0.85	20–30
Spader, medium	0.85–1.2	30–40
Tie tamper	0.6	20
Rammer, foundry, bench type	0.3–0.5	12–16
Rammer, foundry, floor type, medium	0.5–0.6	17–20
Rammer, foundry, floor type, heavy	0.6–0.85	23–30
Rammer, back fill trench type	1	35
Spike driver	1.7	60
Scaling hammer, valveless, for surface work	0.2	7
Scaling hammer for large boiler tubes	0.6	21
Paint scraper	0.2	7
Stone tool, for lettering and light carving	0.2	6
Stone tool, for medium dressing	0.3	10
Stone tool, for roughing and bushing	0.4	14
Weld flux chipper	0.75	24
Pneumatic Tools (Rotary)		
Drilling machine for ¼ in holes in steel	0.3–0.5	10–16
Drilling machine for 3/8 in holes in steel	0.6–0.85	20–30
Drilling machine for ½ in–¾ in holes in steel	0.6–1.15	20–40
Drilling machine for 7/8 in – 1 in holes in steel	1.15–2.3	40–80
Drilling machine for 1¼ in holes in steel	1.25–1.4	45–50
Drilling machine for 1½ in holes in steel	1.7–3.4	60–120
Drilling machine for 2 in holes in steel	2.3–3.4	80–120
Drilling machine for 3 in holes in steel	2.85–3.7	100–130
(For wood boring take next size smaller)		
(For reaming and tapping in steel, take next size larger)		
Corner drills as above		
Grinders with ½ in – ¾ in diameter wheels	0.3–0.6	10–20
Grinders with 2 in diameter wheels	0.3–0.85	10–30
Grinders with 4 in diameter wheels	1.15–1.6	40–55
Grinders with 6 in diameter wheels	1.2–2	45–70
Grinders with 8 in diameter wheels	2–2.5	70–85
Saw, 6 in diameter	0.6	20
Auger drill for coal	2.1	74
Sander (reciprocating or orbital)	0.275	9
Tube cutter, for tubes up to 2½ in	1.25	45
Tube cutter, for tubes up to 2½ in – 4 in	1.6	55
Tube expander, for tubes up to 2½ in	1.8–2.0	65
Tube expander, for tubes up to 3 in	2.125	75
Tube expander, for tubes up to 4 in	2.7	95
Wrenches (rotary torque) for nuts ¼ in	0.15–0.25	5–8
Wrenches (rotary torque) for nuts 3/8 in	4.3–6	15–20
Wrenches (rotary torque) for nuts ½ in – ¾ in	0.75–0.85	25–30
Wrenches (rotary torque) for nuts 7/8 in – 1 in	0.95–1.5	35–40

cont.....

TABLE II – TYPICAL MAXIMUM AIR CONSUMPTION OF PNEUMATIC TOOLS (contd).....

	m^3 free air per minute at 6 bar (gauge)	Cubic feet free air per minute on full throttle 80 lb/in^2 (gauge)
Impact wrench for nuts up to ¾ in	0.75–0.85	25–30
Impact wrench for nuts up to 1¼ in	0.85–1.7	30–60
Spike puller, per spike	0.085	3
Bath enameller	0.25	8
Spray gun, small	0.03–1	1–3
Spray gun, medium	0.1–0.2 at 3.5 bar	3–7 at 50 lb/in^2
Spray gun, large	0.2–0.6 at 3.5 bar	7–20 at 50 lb/in^2
Cement gun, small	2.85	100
Cement gun, medium	4.55	160
Air arbor	0.15	5
Air chuck, as for air arbor	0.004 up	1/8 upwards, per operation according to size
Sump pump, 90–250 gal/m	1.5–3.4	50–120
Air press	0.008 up	¼ upwards per operation according to size
Air motors, up to 1 bhp	0.85–0.9 per bhp	30–35 per bhp
Air motors 1–5 bhp	0.85 per bhp	30 per bhp
Air motors over 5 bhp	0.75 per bhp	25 per bhp
Jigger conveyor engine 11 in	3.4	120
Jigger conveyor engine 13½ in	4.8	170
Jigger conveyor engine 15¾ in	7.1	250
Jigger conveyor engine 17¾ in	10.5	370
Coal cutter, percussive 2½ in	2.8	98
Coal cutter,percussive 3½ in	4.25	144
Coal cutter,percussive 4 in	5.1	180
Coal cutter, Longwall, 50 hp	22.75	800
Drill sharpener, small	1.45	50
Drill sharpener, large	3.4	120
Forging hammer (power hammer) 1 cwt	1.7	60
Forging hammer (power hammer) 3 cwt	3.7	130
Forging hammer (power hammer) 5 cwt	5.4	190
Forging hammer (power hammer) 10 cwt	9.1	320
Forging hammer (power hammer) 20 cwt	15.3	540
Pedestal grinder with wheel, 8 in dia x 1¼ in face	1.3	45
Grouting machine	up to 2.85 at 3 bar	Variable up to 100 ft^3/min at 40 lb/in^2
Rivet forge	0.17	6
Sand blast (shot) 5/16 in jet	2.85	105
Sand blast (shot) 3/8 in jet	4.25 at 60 bar	151 at 60 bar
Sand blast (shot) 7/16 in jet	5.7 at 60 bar	205 at 60 bar
Sand blast (shot) ½ in jet	7.65 at 60 bar	268 at 60 bar
Sand riddler up to 3 ft x 2 ft	0.6	22

cont.....

TABLE II – TYPICAL MAXIMUM AIR CONSUMPTION OF PNEUMATIC TOOLS (contd).....

	m^3 free air per minute at 6 bar (gauge)	Cubic feet free air per minute on full throttle 80 lb/in^2 (gauge)
Squeeze riveter, for 1/8 in x 3/16 in aircraft rivets	0.002 per rivet	1/16 per rivet
Squeeze riveter, for ¼ in cold rivets	0.03–0.1	1–3 per rivet
Squeeze riveter, for 3/8 in cold rivets	0.015 per river	7/8 per rivet
Squeeze riveter, for 7/16 in cold rivets	0.04 per rivet	1½ per rivet
Squeeze riveter (for girder work) for ½ in rivets	0.04 per rivet	1.1/3 per rivet
Squeeze riveter (for girder work) for ¾ in rivets	0.06 per rivet	2 per rivet
Squeeze riveter (for girder work) for 7/8 in rivets	0.08 per rivet	2¾ per rivet
Squeeze riveter (for girder work) for 1 in rivets	0.011 per rivet	4 per rivet
Squeeze riveter (for girder work) for 1¼ in rivets	0.017 per rivet	6 per rivet
Squeeze riveter (for girder work) for 1½ in rivets	0.0275 per rivet	9½ per rivet
Squeeze riveter (boiler work) 1.3/8 in	0.03 per rivet	11½ per rivet
Squeeze riveter (boiler work)1.5/8 in	0.07 per rivet	24½ per rivet
Haulage or winch hoists, pendant wire rope, motor type		3 per foot-ton of lift
Hoists, straight lift, direct acting cylinder type		1½ per foot-ton of lift
Air cylinders		1½ per foot-ton of lift
Stone surfacer, for large blocks		30 per foot-ton of lift
Hydro-pneumatic riveter (as for squeeze riveters)		
Fettling grip or vice	0.004 up	1/8 upwards per operation according to size
Hot miller (1 or 2 cutters)	0.85 per cutter	30 per cutter
Pneumatic drill leg	0.85–0.95	2 above consumption of rock drill
Internal vibrator o/d 2½ in	0.85–0.95	30–35
Internal vibrator o/d 3 in	1.75–2	65–70
Internal vibrator o/d 4½ in	2.4	85
Internal vibrator, o/d 4½ in	1.75	65
Internal vibrator, o/d 5½ in	2.4	85
Shuttering vibrators, 7 lb	0.15–0.3	5–10
Shuttering vibrators, 10 lb	0.3–0.45	10–15
Shuttering vibrators, 14 lb	0.45–0.6	15–20
Concrete compactor	0.17–0.3	6–10

Rotary Tools

Rotary pneumatic tools form a further important group. These again may be heavy duty tools for contractors' or mining applications, or specific workshop tools such as screwdrivers, nut runners, impact wrenches and grinders.

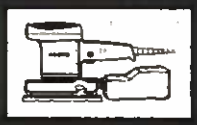
The drive in the case of rotary tools is normally an air motor, driving a work spindle either directly or through reduction gearing. As with percussive tools, the work spindle may be attached permanently to the motor unit, or may be in the form of a detachable fitting. Thus the same basic air tool may be adapted, through various fittings, to power a variety of rotary work heads. It is more usual, however, to find such tools designed as specific rather than as multi-purpose tools.

In the majority of cases, and particularly with smaller rotary tools, vane motors are normally preferred because of their favourable torque/speed characteristics and simple, compact design. Piston motors may be employed for larger tools where higher powers and torque are required at lower operating speeds. Turbine motors are limited in application by their very high running speeds and low torque and are thus used mainly for lightweight ultra high speed grinders and burrs. The large impulse turbine motors with reduction gearing whose performance is competitive with vane or piston motors are, however, coming more into use.

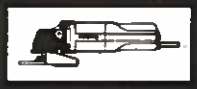
Motor-Powered Tools

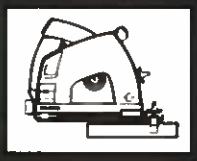
Tools powered specifically by integral air motors, as opposed to rotary tools which are built around an air motor, find their main application in workshop tools, ranging from small, lightweight drills and grinders to large, heavy-duty tools.

Normally all pneumatic tools, regardless of type and size, are designed to operate on conventional compressed air supplies of over 6 bar (80 lb/in^2), with their performance specified accordingly. Typical air consumption figures for a wide range of tools are given in Table II.



air and electric vacuum dust extraction power tools...





FESTO
TOOLTECHNIC

Festo Ltd
Catherine Wheel Road, Brentford, Middlesex TW8 8BB Tel: 01.568.2051 Telex 935705

THOR AIR OPERATED TOOLS

AIR OPERATED TOOLS...

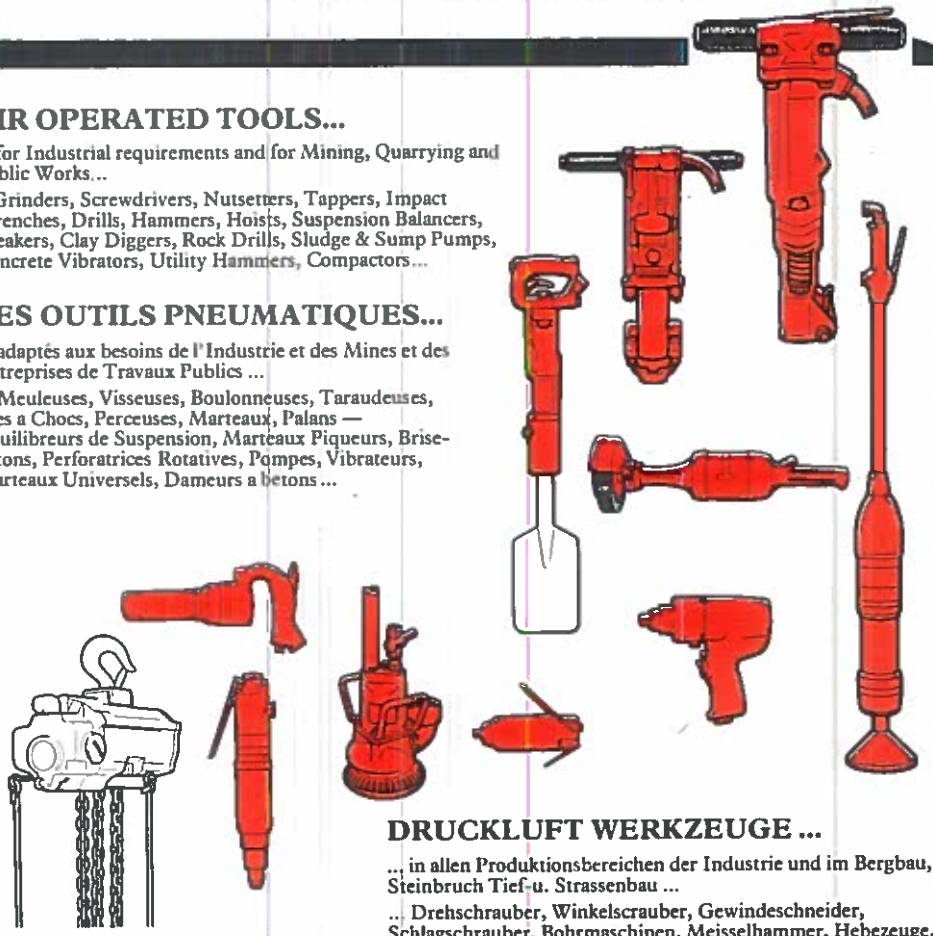
... for Industrial requirements and for Mining, Quarrying and Public Works...

...Grinders, Screwdrivers, Nutsetters, Tappers, Impact Wrenches, Drills, Hammers, Hoists, Suspension Balancers, Breakers, Clay Diggers, Rock Drills, Sludge & Sump Pumps, Concrete Vibrators, Utility Hammers, Compactors...

LES OUTILS PNEUMATIQUES...

... adaptés aux besoins de l'Industrie et des Mines et des Entreprises de Travaux Publics ...

... Meuleuses, Visseuses, Boulonneuses, Taraudeuses, Cles a Chocs, Perceuses, Marteaux, Palans — Equilibreurs de Suspension, Marteaux Piqueurs, Brise-betons, Perforatrices Rotatives, Pompes, Vibrateurs, Marteaux Universels, Dameurs a betons...



DRUCKLUFT WERKZEUGE ...

..., in allen Produktionsbereichen der Industrie und im Bergbau, Steinbruch Tief-u. Strassenbau ...

... Drehschrauber, Winkelscrauber, Gewindeschneider, Schlagschrauber, Bohrmaschinen, Meisselhammer, Hebezeuge, Federzuge, Hammer, Abrau- und Spaten Hammer, Gesteins Bohrhammer, Pumpen, Ruttler, Mehrzweckhammer, Kompaktor-Stampfer ...

SYMBOL OF SW EXCELLENCE

STEWART-WARNER LTD
THOR TOOLS DIVISION
Tynemouth, Tyne & Wear, England. Tel: North Shields (0632) 590655 Telex: 53602. Answer Back THORSW G

Contractors' Tools

THE MAJORITY of tools in this category come under the general heading of breakers. They are basically hammers used in combination with chisels or steels for demolition or cutting on various materials. The shape of the steel will depend on the material to be worked, a pointed steel being used on concrete and similar hard materials, and a wide chisel-edge for softer materials such as asphalt. For digging or trenching work a spade-shaped steel can be fitted, whilst further attachments may adapt the breaker for ramming, pile-driving, *etc*.

Tools of this type normally range in weight from about 14 to 40 kg (30 to 90 lb), depending on the specific duty intended which also influences the degree of robustness of construction. Arbitrary classifications are 'light', 'medium' and 'heavy' duty.

Heavy breakers are used for demolishing concrete foundations and walls, breaking concrete pavement, trenching in very hard ground, *etc*. Medium breakers are used for breaking light concrete pavements and floors, tarmac and similar surfaces. Light breakers are used for light demolition work on floors, paving and masonry walls, and as general utility 'trimming' tools.

Air consumption in the case of breakers will vary considerably with size and duty, and may range from 0.93 m^3/min (30 ft^3/min) at 6 bar (80 lb/in^2) for light breakers to 2 m^3/min (70 ft^3/min) at 6 bar (80 lb/in^2) for heavier tools.

All breakers employ solid steels which are driven without rotation and thus provide, in effect, a straightforward chiseling action for breaking, driving or tamping. A wide variety of different tools may be used, depending on the work involved. Fig 1 shows a selection of steels used for road ripping. Further attachments can include pipe or sheeting drivers, spike drivers, light pile drivers, *etc*, depending on the capability of the tool – *eg* sheeting and light pile driving would normally be limited to heavy breakers.

Various types of tool retainers are used such as a latch or spring as well as link locks and patented retainers. The size of the tool shank varies according to the class of the breaker, *eg* from 25 mm by 100 mm (1 in by 4¼ in) in the case of light breakers to 30 mm by 150 mm (1¼ in by 6 in) for heavy breakers.

In the case of breakers, a T-shaped handle is normally employed with the throttle valve conveniently located above one side of the handle. The throttle action normally also operates a built-in lubricator as adequate internal lubrication is an essential feature with such tools. This does not necessarily preclude the use of an air line lubricator. In fact, the additional use of an air line lubricator is generally recommended.

Fig 1 Road ripper accessories. Top row (left to right) chisel edge steel (asphalt cutting), wide curved scoop, wide flat spade, digger. Bottom row (left to right) plug and feathers, tamper, chisel pointed steel, frost wedge, pointed steel.

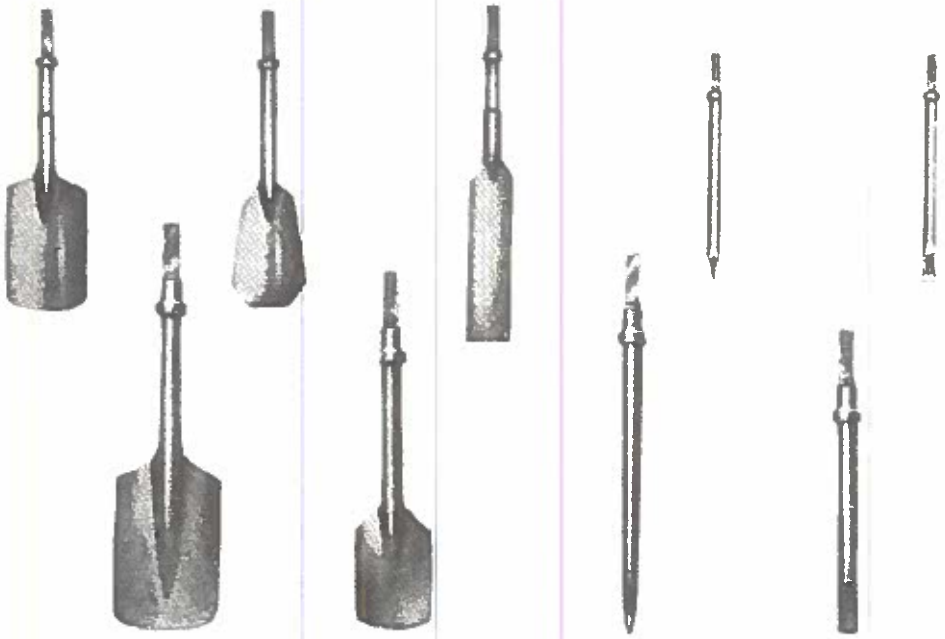


Fig 2 Range of accessories. Top row (left to right) curved scoop, flat spade, digger blade, moil point, chisel point. Bottom row (left to right) curved scoops, moil point, chisel point. (Holman Bros Ltd).

Spaders (Diggers)

When designed specifically as a spader or clay digger the tool is normally smaller than a concrete breaker and generally has a D-shaped handle. Provision may also be made to fit an extension handle so that the tool may be manipulated and controlled when digging narrow trenches. All types of spaders consist essentially of a light to medium duty size hammer body capable of taking a variety of different spade, scoop and chisel shaped steels. Air consumption of a typical light duty spader is usually of the order of 0.7–0.85 m^3/min (25–30 ft^3/min) at 6 bar (80 lb/in^2) and that of a medium duty spader 0.85–1.13 m^3/min (30–40 ft^3/min) at 6 bar (80 lb/in^2).

A spader may also be used for light demolition work, or as a pick, by fitting a moil or chisel point instead of a spade – see Fig 2. It may also be produced in specific forms, viz:

Clay digger – for digging or trenchwork, fitted with a D-shaped handle and inside or outside throttle control trigger.

Trench digger (Trencher) – with an extended T-shaped handle for open trench evacuation, the length of the handle allowing the operator to stand upright.

Tampers

A tamper is a pneumatic hammer fitted with a tamping bar. Size and weight can vary considerably, but a typical tool of this class would weigh some 13.8–18 kg (30–40 lb) with an air consumption of the order of 0.57 m^3/min (20 ft^3/min) at 6 bar (80 lb/in^2).

Special designs of tamper may be used for work in confined areas, also multiple tampers for working over large unrestricted areas. Thus a duplex tamper would comprise two separate tampers, parallel mounted with a common handle, and a triplex tamper, three separate tampers similarly mounted. The triplex arrangement is much to be preferred for multiple operation since the tamper blows can be desynchronized to produce denser compaction.

Rammers

A rammer is essentially the same tool used for such work as consolidating back-fill in trenches and excavations and general compacting work which may come under a contractor's specifications. The typical contractors' tool of this type has an air consumption of the order of 1 m^3/min (35 ft^3/min) at 6 bar (80 lb/in^2). Smaller rammers, with correspondingly reduced air consumption, are used for foundry work (see chapter on *Workshop Tools).* Large rammers may be trolley mounted for ease of manipulation.

Concrete Vibrators

Light, fast-acting pneumatic hammers are used for concrete vibration by being bolted or clamped to the shuttering. They are normally relatively small in size (*eg* about 4.5 kg (10 lb) weight) and any number of hammers may be attached to the shuttering, depending on the thickness and surface area of concrete involved. Rammers also are sometimes used for concrete vibration although they are less effective than vibrators for this class of work since the latter may have speeds of up to 5 000 blows per minute.

Stone-Working Tools

Special stone-working tools for light or medium dressing range in weight up to about 2.7 kg (6 lb) with air consumption figures from 0.7–0.85 m^3/min (5–10 ft^3/min) at 6 bar (80 lb/in^2). They are normally high-speed hammers (often of valveless construction) and the tool tip may be permanently attached and hardfaced, or detachable. Slightly larger tools of this type may be used for roughing and bushing, with air consumption figures of up to 0.42 m^3/min (15 ft^3/min) at 6 bar (80 lb/in^2).

In addition, contractors' work may involve the use of tools more properly classified under mining tools, or rock drills.

Rock Drills (Figs 3 and 4)

As a rough classification, portable or handheld rock drills can again be grouped as 'light', 'medium' and 'heavy', typical weights being 16 kg (35 lb), 20 kg (45 lb) and 25 kg (55 lb) respectively. Heavier rock drills of 34 kg (75 lb) weight or more are sometimes referred to as 'sinkers', particularly when employed for shaft sinking.

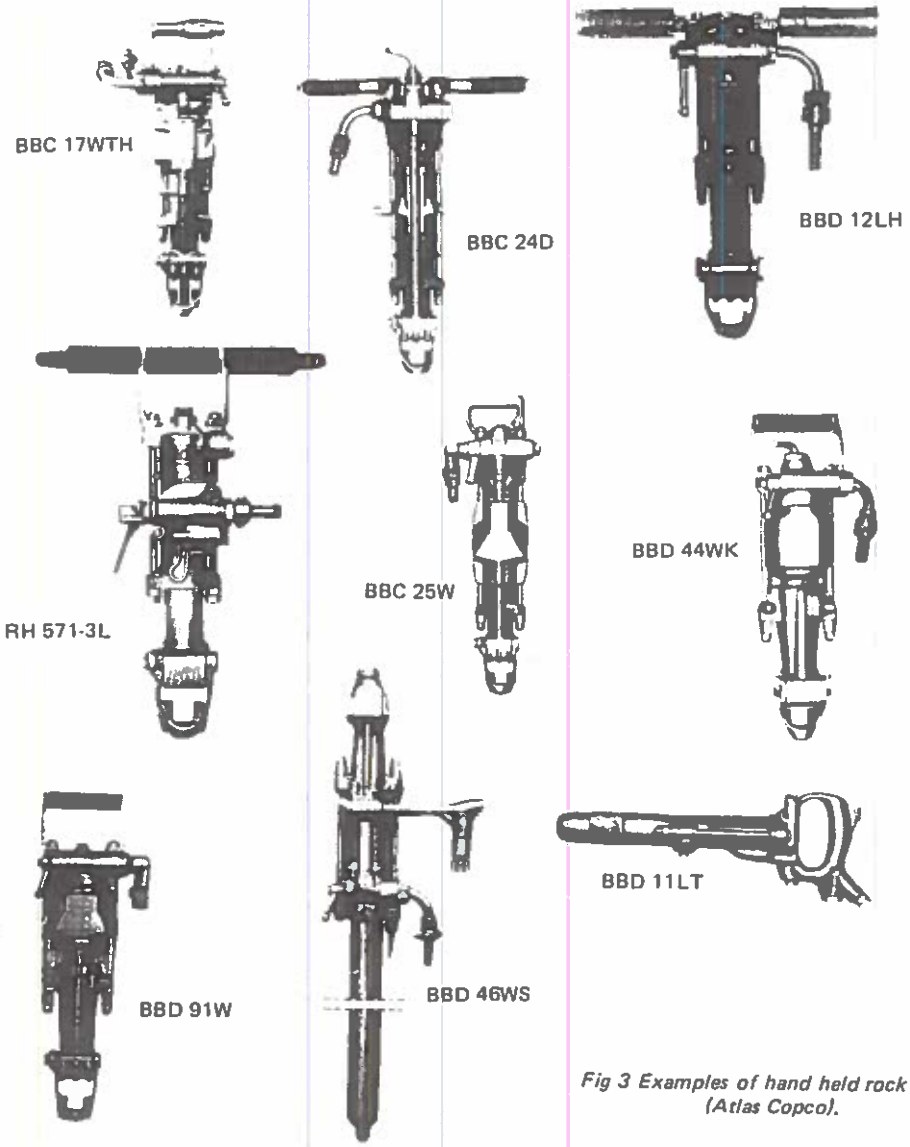


Fig 3 Examples of hand held rock drills. (Atlas Copco).

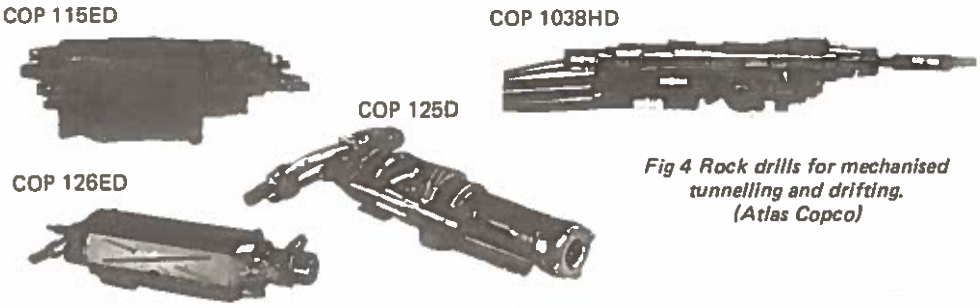


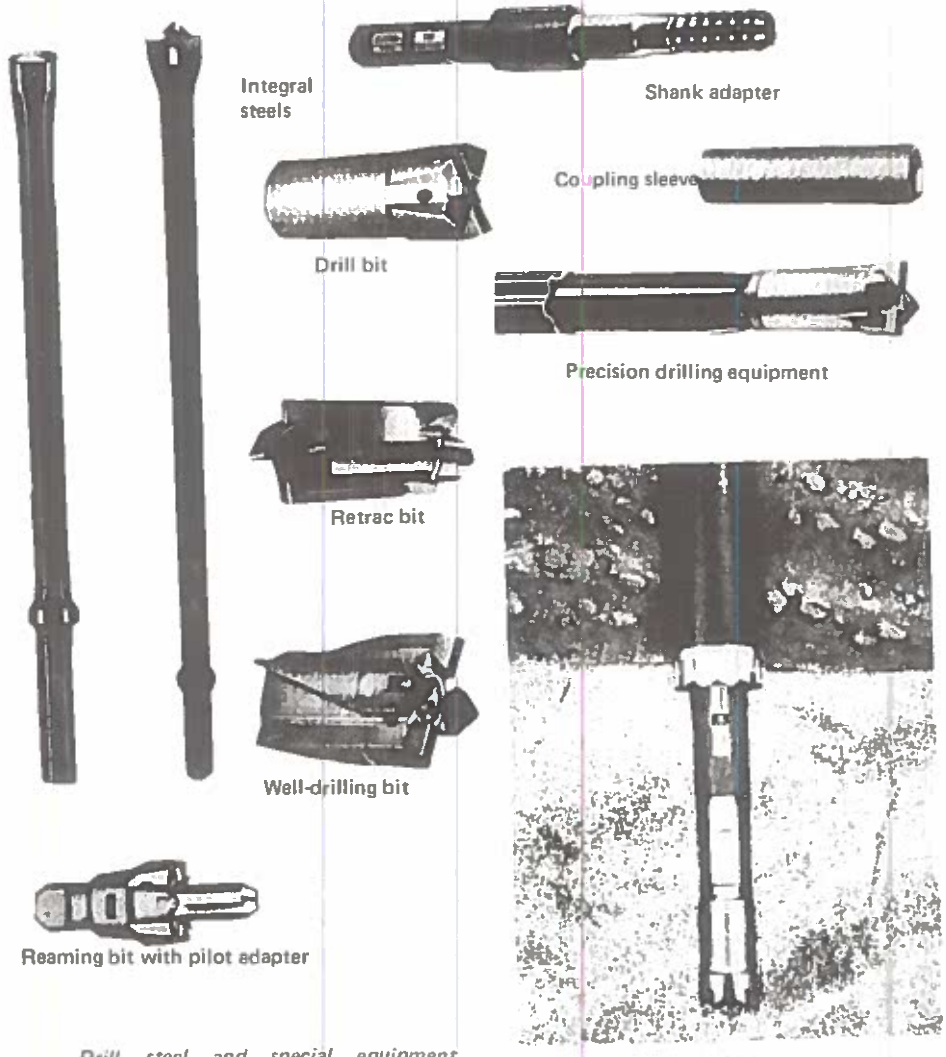
Fig 4 Rock drills for mechanised tunnelling and drifting. (Atlas Copco)

A rock drill is a percussive hammer driving a drill bit. Although the form and construction is similar to that of a concrete breaker, one difference is that on the upstroke spiral flutes on a twist bar or rifle bar engage with a nut on the top of the piston causing the drill bit to be rotated through part of a turn. The other end of the rifle bar accommodates pawls operating in a ratchet ring, disengaging on the downstroke to allow the drill to be driven straight without rotating. Each return stroke (only) is thus accompanied by partial rotation of the drill steel so that the resulting hole is circular. The rate of rotation of the drill is arranged to suit the conditions under which it will be used and some designs may incorporate provision for adjusting the rate of revolution. Other designs may incorporate a control releasing the ratchet so that starting and stopping of drill rotation is under the control of the operator. In the case of very heavy duty drills — *eg* large capacity drills of large diameter — rotation of the drill may be achieved by a separate air motor (usually piston type) because of the high torque required to rotate the drill.

Rock drills may also be classified as 'wet' or 'dry' types depending on the method employed for cleaning the drill hole. 'Wet' drills are normally used for underground mining where the drilling dust is damped down by a water spray applied under pressure from an outside source led down to the bottom of the hole through an internal water tube. Dry drills utilize compressed air for hole cleaning, by either continuous blowing or a pulsating flow of compressed air supplied through the steel during drilling — see also Table I.

TABLE I — ROCK DRILL CATEGORIES

Category	Plug	Light	Medium	Heavy	Sinkers
Weight	9 kg (20 lb)	16 kg (35 lb)	20 kg (45 lb)	25 kg (55 lb)	34 kg or more (75 lb or more)
Duty	Plug and feather work light hole drilling	Coal mining, small hole drilling	General drilling work	Deep rock drilling	Shaft sinking
Depth		1.8–2.4 m (6–8 ft)	5–6 m (up to 16–20 ft)	7.6 m (up to 25 ft)	7.6 m (up to 25 ft)



Drill steel and special equipment. (Atlas Copco).

It will be appreciated that although it is capable of cutting a circular hole the actual action of a rock drill is that of continual hammer blows crushing the rock. To keep the hole free and reduce the risk of the drill jamming, compressed air or water can be forced into the hole through a hollow drill, blowing or washing away the crushed rock parts as they are produced.

In the case of compressed air blowing the operator has various methods of directing compressed air to the bit end, such as opening a transfer valve to limit piston travel and/or direct air to the bit end, or holding the piston at mid stroke to allow live air to travel down to the drill shank, *etc.*

With wet drilling — also known as dustless drilling since the water fed into the hole damps down the dust formed by the drilling action — the water stream may be led down through an internal water tube. A separate valve controls the water flow and this may be used with compressed air blowing — *ie* cleaning of the air is done by air blowing, and damping of the dust by the water stream. An alternative method still used is an attachment fitted to the side of the drill which directs an atomized spray of water onto the drill and into the hole. All have certain limitations as regards true 'dustless' drilling.

Drifters

A drifter is a rock drill specially adapted for horizontal boring, mining or tunnel driving, *etc,* although it may also be suitable for angle drilling. It may be supported on a column or crossarm, or fitted to jibs or booms for enhanced mobility, in which case a number of drills may be mounted on the same chassis or trolley. Multiple drilling rigs of this type have particular application for quarry work as well as tunnelling and underground working. Standard drifters are made for wet or dry drilling with controlled blowing, or in vented form.

Stopers

Stopers are used mainly for overhead vertical or near-vertical drilling in underground workings and comprise a percussive-type air drill attached to a pneumatic cylinder which acts as a telescopic leg both to support the drill at the working height required and to provide forward feed. Modern types of stoper may also be provided with automatic self-rotation in a similar manner to that standardized for rock drills (*ie* by means of a rifle bar and ratchet mechanism) although the design differs in other details (*eg* stopers may employ a collared chuck for retaining the steel and omit the usual anvil action for increased speed of drilling). In other respects they are similar to rock drills and are made for both wet and dry drilling.

Self-Propelled Drills

In mining and quarrying, considerable use is made of self-propelled drifters and similar drills, and such tools can also be mounted on a wheeled cradle or trailer which can be pushed or towed to the position required. The drill itself is mounted on a boom and controls over drill position, angle and rate of feed may be manual or provided by pneumatic or hydraulic power.

Fully mobile, self-propelled units may or may not incorporate a compressor as part of the rig and may be based on a wheeled or crawler tractor configuration. In such larger units hydraulic power is often employed for raising and lowering the boom, the hydraulic pump being driven by the engine; or all motions may be operated entirely from a compressed air supply. Drilling rigs of this type may also be designed in the form of attachments for mounting on standard tractors, with provision made for both percussive and rotary drilling by fitting an adaptor and rotary bit in place of the hammer unit.

Rotary Tools

In the case of a rotary tool the driver is in the form of an air motor built into the tool, the high power/weight ratio of such motors making them particularly adaptable to maintaining speed under load. Free running speed as well as load speed can be controlled through the use of a governor or similar device, thus enabling constant speed working to be maintained, if necessary under varying load. This is particularly applicable in the case of such tools as grinders and rotary drills. Gear trains and clutches may be added to all types of motor, with the torque multiplication obtained

by reduction gearing especially significant in the case of torque wrenches, nut setters, *etc* — tools subject to high torque loading.

Rotary tools may be grouped in the same way as percussive tools, and individual tools are discussed under their respective headings.

Rotary Contractors' Tools

There are considerably fewer rotary tools in this category than percussive ones, although a variety of types have been developed for specific applications. These embrace, mainly, concrete vibrators, shank grinders, sump pumps, hoists and winches (although the last two are more properly classified as miscellaneous pneumatic devices).

Concrete Vibrators

Rotary type concrete vibrators are normally on the 'internal' type for placing within the mass of concrete — *ie* the concrete is dumped over the machine to completely immerse it or the vibrator so designed that it can readily be immersed in a mass of concrete. The concrete mass is then vibrated by operating the machine for a predetermined period of time (usually about half a minute or slightly less).

The basis of a concrete vibrator is usually an eccentric weight mounted on low friction bearings and capable of being rotated at high speed by an air motor. The unit is fully enclosed within a steel casing which is completely sealed, thus preventing entry of either water or concrete. Both delivery and exhaust (air return) lines are attached to the casing so that exhausted air is carried away from the machine and not vented into the concrete mass. Sizes vary from about 62.5 mm to 160 mm (2½ in to 6 in) diameter, with weights between about 11.5 and 45.4 kg (25 and 100 lb).

A vane type motor is normally employed for this type of tool with the speed limited by a centrifugal governor linked to the air control valve.

Shank Grinders

The shank grinder is a form of rotary grinder with a specially shaped recessed cup grinding wheel for trueing, squaring up and deburring the ends of steels used on percussive type tools. The wheel is normally driven by a vane motor fitted with a centrifugal governor and provision is made to oscillate the wheel across the workpiece, *eg* by employing a pivoted grinding head manipulated by a hand lever. Other controls may be provided for feeding and clamping the steel or workpiece in position. The centre of the wheel may also be fitted with a reaming bit.

Sump Pumps (Figs 5 and 6)

Sump pumps are designed expressly for dewatering excavations, *etc*, where they may be called upon to handle sludge and mud as well as clear water. It is also desirable that the geometry of the unit be arranged to pump down to the lowest possible level.

A sump pump thus consists of a suitable pump (almost invariably a simple self-priming centrifugal type) powered by a suitable driver (an air motor in the case of a pneumatic unit). Both piston and vane type motors may be used, although the latter are usually preferred. The motor spindle is then usually connected directly to the pump at motor speed.

Speed control is accomplished by a centrifugal governor or similar device, control being necessary to prevent the motor (and pump) overspeeding should it be left running after pumping dry. Equally, the pump must be suitable for running 'on snore' without damage, unless provision is made for automatic cut-off and restart.

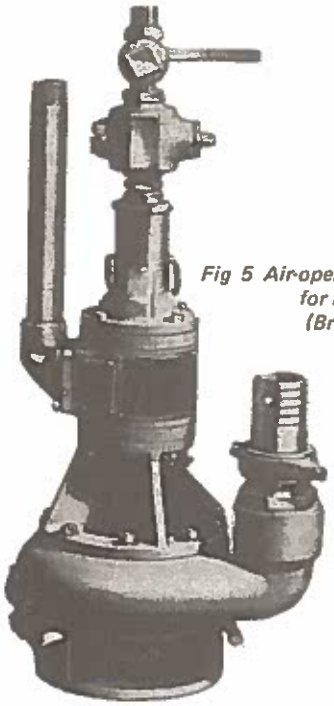


Fig 5 Air-operated sump pump type 38 for industrial duties. (BroomWade Ltd).

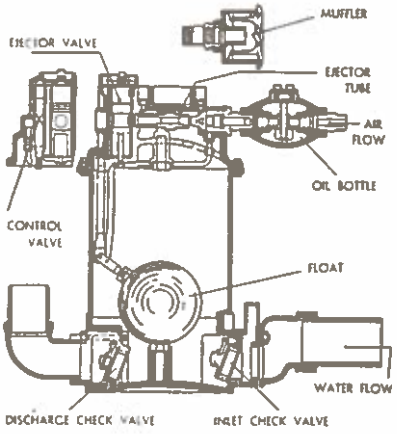


Fig 6 Ejector type sludge pump. (Consolidated Pneumatic Tool Co.Ltd).

The compressed air sump pump has the additional advantage that where high heads are involved delivery can be assisted by air lift by injecting the air stream into the pump outlet. In this manner greater heads can be achieved than those delivered by the pump alone. This is not necessarily a feature of sump pumps as such, but may be incorporated on units designed for high heads.

Workshop Tools

AIR-OPERATED workshop tools can be divided handily into rotary tools generally powered by air motors and percussive tools operating on a similar principle to rock drills, etc.

Rotary tools can be divided under the following headings:-

(i) grinders and abrasive tools;

(ii) drills;

(iii) screwdrivers, nut runners, impact wrenches, etc.

There are also a considerable number of specialist tools developed for particular industries and standard types of air-operated tools adapted to specific duties.

Percussive tools such as chipping hammers, rammers, core busters, riveting hammers, etc, are all basically the same type of tool, but with differences in stroke and diameter of piston and speed, according to the duty specified. Scaling hammers are designed for fast speeds and low impact energy blows. Chipping hammers have a high speed with a medium or short stroke. Riveting hammers normally have a long stroke and a slower speed of working.

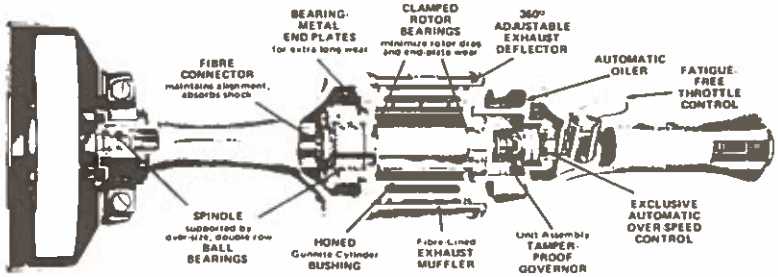
All such hammers normally have a throttle valve regulating the air supply so that the hammer may be started slowly if desired, and subsequently operated at any speed from slow to maximum. The return stroke of the piston is cushioned to make the tool easier to hold and manipulate.

Grinders and Abrasive Tools

These, basically, comprise an air motor carrying a grinding wheel or abrasive wheel, etc, normally on the shaft extension, or in some cases driving the wheel through reduction gearing. The fitting of buffing wheels, wire brushes, etc, is usually a simple alternative, and thus the basic grinder becomes a multipurpose abrasive tool in this respect. Grinders may be made as portable or fixed units, depending on size and duty requirements. Portable units predominate numerically.

Most modern grinders are fitted with governors to ensure an optimum operating wheel speed without exceeding the maximum.

Vitrified bonded wheels are limited to a maximum peripheral speed of 6 500 ft/min (33 m/s) permitted speed when idling. Conventional resin-bonded grinding wheels permit maximum peripheral speeds up to 9 500 ft/min (48 m/s). Glass-reinforced grinding wheels and specially sectioned wheels may be capable of safe operation at peripheral speeds up to 16 000 ft/min (80 m/s).



Straight grinder shown in section (Thor Tools).

Grinders may be generally classified as –

(a) straight or horizontal grinders

(b) vertical or surface grinders

(c) angled or angle-headed grinders, referring to the attitude of the grinding wheel relative to the body in portable tools, and

(d) pedestal grinders where the complete tool is mounted as a fixed unit, normally with foot control of the motor to permit the workpiece to be manipulated with both hands.

Other types of 'fixed' grinder for bench mounting are basically portable grinders mounted in suitable stands.

Sizes of portable grinders range from very small, light, high-speed units running up to 100 000 rev/min (in which case turbine motors may be used) and with a maximum wheel capacity of about 2 in (50 mm) diameter, to large, heavy-duty models running at up to 4 000 rev/min with a wheel capacity of up to 8 in (200 mm) diameter. There is a very wide variety in individual detail design, and also in the production of 'specific type' grinders – eg for particular types of dressing work.

Die-grinders for small wheel sizes may employ vane type motors for speeds up to 40 000 rev/min although 20 000 rev/min is the more usual maximum when a wheel is fitted. With the high operating speeds burrs, rotary files and abrasive points are the normal attachments. Typical duties for this class of tool include deburring, touching up welds, finishing castings, etc. The very light point grinders with speeds of up to 70 000 rev/min employ turbine motors.

Conventional grinders, or 'straight' grinders are normally classified by their wheel capacity (ie grinding wheel size) related to the maximum permissible or 'safe' speed for the wheel. Speed control is provided by a governor to restrict the air supply as load decreases, and vice versa. By changing the tension of the governor spring or the size of the governor weights the speed of the motor can be altered to suit various types or sizes of wheel. Speed governors also restrict the no-load air consumption of the motor and safeguard the vanes and cylinder against undue wear.

Sanders

Vertical sanders are essentially the same as vertical grinders with the exception that a sanding disc and pad is substituted for the grinding wheel. Standard disc sizes are 5 in (125 mm), 7 in (175 mm) and 9 in (225 mm) diameter. Standard speeds normally range from 1 000 to 10 000 rev/min, depending on the size of the tool and the duty involved.

For lighter work the right angle sander is particularly attractive since the tool body and handle are at right angles to the disc. This may take the form of a right angled attachment for fitting to a

normal grinder body or a right angled drive incorporated as an integral feature of the body, with or without reduction gearing. Speeds normally range from 2 000 to 10 000 rev/min, depending on the size and duty.

Horizontal grinders may be adapted as belt sanders (or belt grinders) by the fitting of an attachment comprising, basically, a large diameter driving pulley separated from a jockey wheel with an abrasive belt looped between the two wheels. The driver wheel is also used as the contact wheel for working and is commonly made of rubber.

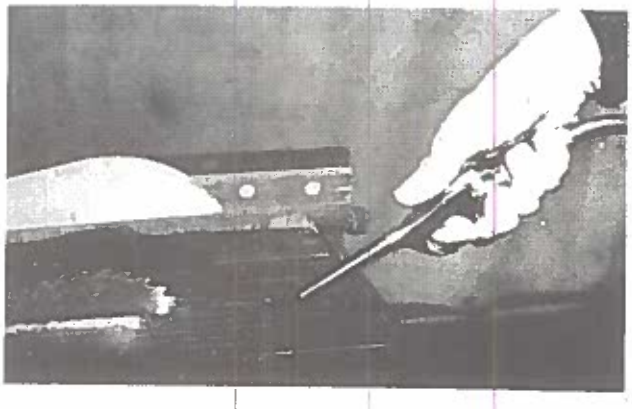
Orbital sander (Sundstrand)

In addition to disc and belt sanders, portable air-operated orbital sanders are produced for both wet and dry sanding operations. A typical tool of this type weighs 4–5 lb (1.8–2.25 kg) with an air consumption of 6–10 ft^3/min (0.17–0.28 m^3/min) and an oscillating rate of 5 500 cycles per minute, and is often designed for moderate air pressures (eg 50–75 lb/in^2) (3.5 to 5.25 bar). Larger machines with opposed pads reciprocating in opposite directions are also available for production line work, etc.

Rotary Files

Rotary files and burrs are used for light finishing work, weld finishing, deburring, etc. Spindle speeds are somewhat lower than that common with grinders, from 1 000 to 3 000 rev/min being usual when dealing with hard metals. Higher speeds can, however, be used on softer materials and also when using carbide burrs.

For small and intricate work the burr or rotary file may be driven via a flexible extension shaft for maximum ease of manipulation, and also to permit access of the tool into restricted areas. Speeds are usually higher with small diameter burrs, and in specialized applications, such as dental drills, the tool may be powered by a turbine motor.



Blow gun (Dyna-Quip Ltd)

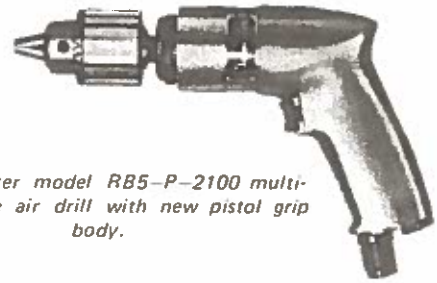
Dust Extractors

Dust extractors may be fitted as separate devices to grinders, sanders, wire brush machines, fettling chisels, etc, to extract at source dust created by the tool. They are normally designed specifically to suit a particular type of tool – eg grinding wheel, sanding disc, chisel, and comprise, basically, a suction nozzle or extractor head connected by a flexible line to the extractor.

Drills

Sizes range from the smallest lightweight drills weighing less than 1 lb (0.45 kg) with a drill capacity of up to 1/8 in (3 mm) diameter, to large, heavy duty tools which may weigh up to 150 lb (68 kg) and have a drill capacity of 3 in (75 mm) diameter. Free-running speeds may vary from 15 000 rev/min in the case of the small light duty drills down to 75–125 rev/min in the case of heavy-duty tools. Speed control is usually provided via a throttle valve and most types are available either with uni-directional or reversible rota ion, the latter being a necessary feature for tapping and other duties. Non-reversible drills may be preferred for specific operations, eg reaming.

Angle drill (Thor Tools).



Desoutter model RB5–P–2100 multi-purpose air drill with new pistol grip body.

Although described as drills, tools of this type can take a variety of bits other than drills, and thus may be equally suited for reaming, tapping, stud setting, stud driving, tube rolling, etc. Many tools so described are, in fact, air drills with the appropriate attachment. Similarly, other attachments may be available for drilling, etc, with fixed angle heads, 30°, 45° and 90°, and adjustable angle heads. The shape may vary from purely cylindrical form to pistol grip tools, either or both capable of taking a variety of attachments.

Performance of air drills, particularly in the lightweight sizes, is generally superior to any other type for virtually all drilling jobs, with the lightweight and high power/weight ratio making for faster working and easier handling in the case of portable tools. Air drills can also be readily adapted to self-feed or mechanical feed, either as portable units or mounted in fixtures, as well as being particularly suited to multi-drilling rigs.

Larger heavy-duty air drills are normally fitted with a Morse taper socket chuck, sizes ranging from No 1 or No 2 in the case of 3/16 in (14 mm) and 7/8 in (22 mm) drills up to No 5 in the case of 3 in (75 mm) capacity drills. The chuck may also be fitted with an ejector pin to enable the drill to be removed readily. Large capacity drills also normally incorporate some form of self-feed.

Wood Borers

Wood borers, again, are basically air drills designed for slower running speeds and fitted with reversible motors to assist in backing off and withdrawal of the auger bit. Typical size ranges and speeds are:

up to 1 in (25 mm) auger capacity, 750 – 1 000 rev/min
up to 2 in (50 mm) auger capacity, 600 – 750 rev/min
up to 3 in (75 mm) auger capacity, 250 – 350 rev/min

Tools in this category are widely used for heavy timber construction. For lighter wood boring duties standard air drills may be used with auger bits, driven through a reduction gearbox if necessary.

Woodworking

The range of pneumatic powered tools for timber and woodworking duplicates that of electric portable tools, with certain advantages. For example, it is readily possible to provide reversible motors in air-powered tools (eg woodborers). In addition, portable pneumatic tools are light in weight for their work capacity, generally reliable in operation, safe to use under conditions where an electric motor to electric supply may represent a hazard, and have low operating costs. Also an air-powered tool can be stalled without damage, unlike an electric powered tool.

Woodborers – these are designed specifically for accommodating a range of auger bits with lower rotational speeds than drills. The usual capacity range is from ¼ in (6.4 mm) up to 4 in (100 mm) diameter (auger size) and in various sizes of shank. Various auger shapes are used, according to the timber to be bored – eg a Scotch auger for rough timbers, a round nose auger for angular boring, special augers for green-heart, etc. The larger woodborers are usually reversible as this is a highly desirable feature where large holes are being drilled.

Drills – the pneumatic drill is the counterpart of the electric drill and for woodworking may be fitted with a wood bit chuck. Working capacity usually ranges up to about 7/8 in (22 mm) auger diameter, according to the size of the tool. The drill may or may not be reversible. In the latter case it is generally recommended that the auger diameter be restricted to a somewhat lower figure, especially when long augers are being used.

Saws – tools available in this class range from pneumatic-powered chain saws for contractors' work to small portable or bench type circular saws. Reciprocating saws are also produced, usually in smaller sizes for fitting with standard hacksaw blades. A particular feature of a pneumatic powered chain saw is that it can be used under water. In this case a safety throttle is advisable so that the saw motor stops immediately should the driver(operator) accidentally lose his grip on the tool.

Adze – for use as an adze a timber slicer can be fitted to certain types of pneumatic hand drills. The light overall weight of the resulting tool makes it particularly easy to manipulate compared with any other form of powered adze.

Screwdriving, nail driving, sanding and similar duties can be also be performed by specific and standard pneumatic tools, the range of such tools and attachments being very wide.

Pneumatic screwdrivers (Atlas Copco)

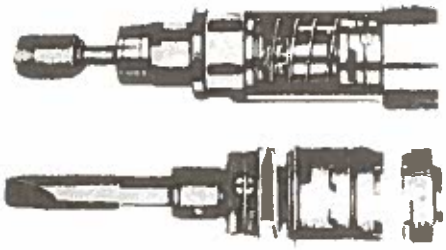
Screwdrivers

Screwdrivers are normally powered by vane motors driving through planetary reduction gearing to reduce the spindle speed to a suitable value. The motor may or may not be reversible. There is considerable variety in detail design and shape – eg straight or offset handles. Although they are

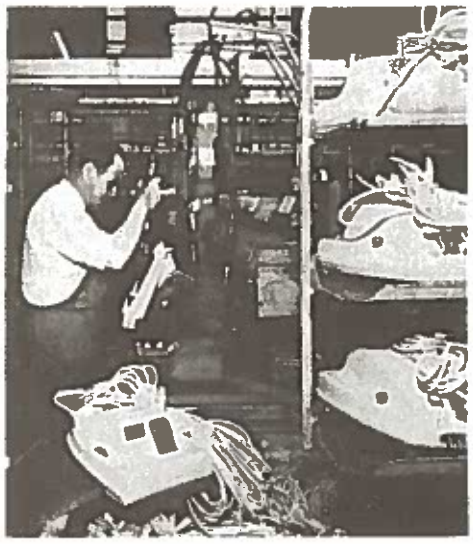
light and compact, provision may also be made for suspension of a tool by lightweight balancers to reduce operator fatigue on a production run, whilst other devices such as pneumatic or magnetic pick up of screws may be incorporated. Numerous clutch and torque control methods are also employed.

With geared drives speed and torque are directly related, ie a low speed gives higher torque, and vice versa. Choice of torque and speed depends on the nature of the assembly, the materials involved and the speed and accuracy required. In some designs of screwdriver there is provision to adjust torque. Ideally the choice should be a screwdriver which is fast, easy to handle, and adaptable to different jobs (unless employed at a specific point on a production line where the duty requirement does not change).

With direct drive the screwdriver bit is connected directly to the motor via the reduction gearing, hence maximum torque is equal to the stall torque of the motor multiplied by the reduction gearing, the air pressure being regulated so that the tool will stall when the screw is driven home. A simple clutch may, however, be fitted in order to make it possible for the operator to present the tool to the screw with the motor running, although the driver is still classified as a direct drive type since there is no limitation on applied torque below the stall torque.



Uni-tork (top) and adjustable clutches for screwdrivers/nut-setters (Thor Tools).



Desoutter screwdrivers in use on a vacuum cleaner assembly line.

Where torque limiting is required, either a positive clutch or a torque-limiting clutch can be incorporated. In the former case the clutch is, usually, of the dog type held apart by a spring, the clutch being engaged by forward pressure on the tool. Releasing pressure on the work allows the clutch to disengage, thus the motor keeps running although the screw is not being driven home. In practice this means that the operator can regulate the amount of torque transmitted by varying the amount of downward pressure applied to the tool.

With a torque-limiting device the clutch is a little more complicated but can be preset to slip at a given torque level. The tool will then transmit torque only up to that level, after which the drive will slip. There are numerous designs of torque-limiting clutch which provide this facility, the

most common usually employing two sets of opposing clutch faces, one set providing the engaging clutch and the other the slipping clutch disengaging at a predetermined torque level.

Screwdrivers with torque-limiting clutches are generally to be preferred, particularly for light tools used for work on finished surfaces or where precise fastening torques are required, as they are far less liable to slip or damage the fastener and do not place a premium on operator skill.

Nut Setters

Basically these are identical to screwdrivers with a socket bit replacing the screwdriver bit. They are produced in straight or angled forms and with a handle or grip shape and size consistent with the working loads and reactions likely to be involved.

Torque control may be even more important in order to avoid over-tightening, in which case a torque-limiting clutch is to be preferred. Motors are invariably made reversible to enable the runner to be used for assembling and removing nuts. Larger sizes of nut runners need not be fitted with a clutch since in such cases the operator can more readily handle the stall torque of the tool without losing control.

The inherent characteristics of both screwdrivers and nut setters make them particularly suited for multi-point working assembled on a suitable jig or fixture for production line work. Close spacing can be achieved by stacking the individual tools at different levels or positions, as necessary, with extension bits to bring all the sockets into the same plane as the work. In multiple units it is also possible to arrange for a pre-set sequence of tightening individual nuts if required.

Reamers

Special tools are not normally produced for reaming since this duty can be performed by a standard drill reducing the capacity by approximately 25%, ie the reaming capacity can be taken as 75% of the drill capacity.

Virtually any size of drill can be used for reaming with such a reduction in capacity. Smaller drill sizes, however, may exclude tapping as a permissible duty.

Tappers

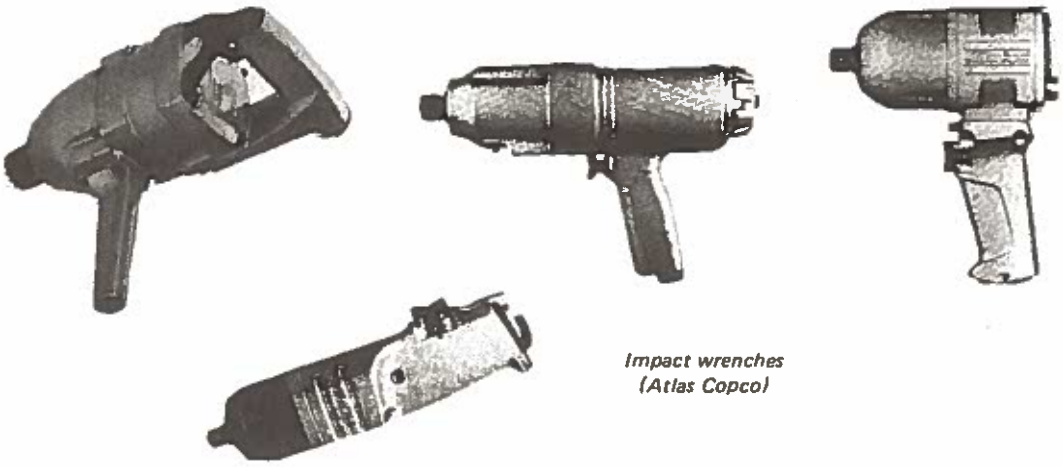
Standard air drills, nut setters, screwdrivers, etc, may be used as tappers within the capacity of the motor, which will be lower than the capacity for drilling. The reduction in capacity is usually of the order of 40–20% – eg typically a drill with a drilling capacity of 9/16 in (14 mm) will have a tapping capacity of 5/16 in (8 mm) to 7/16 in (11 mm), a size with a 1 in (25 mm) drilling capacity would have a tapping capacity of 3/4 in (20 mm), and so on.

In addition, air tools may be specifically designed as tappers, when it is usual to provide a higher reverse speed for withdrawal of the tool.

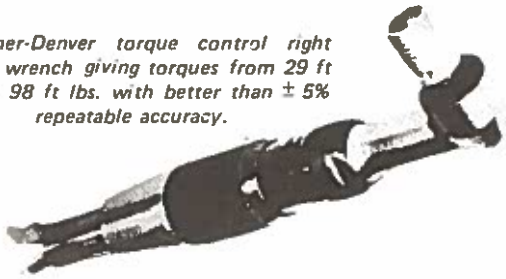
Impact Wrenches

The impact wrench is merely another form of the screwdriver or nut setter except that with increasing torque load the driving or running action is automatically converted from a smooth torque into a series of rapidly repeated impacts. Besides making it possible to exert very high tightening (or loosening) torques (very much higher than those provided by a continuous drive), torque reaction on the operator is practically eliminated since the turning effort at this stage is developed by impact blows instead of sustained torque.

Typical sizes of impact wrenches range from about 1½ lb (0.7 kg) in weight with a bolt capacity of 7/32 in (5.5 mm) diameter, up to 600 lb (270 kg) with a bolt capacity of 12 in (300 mm) diameter.



Impact wrenches (Atlas Copco)



Gardner-Denver torque control right angle wrench giving torques from 29 ft lbs – 98 ft lbs. with better than ± 5% repeatable accuracy.

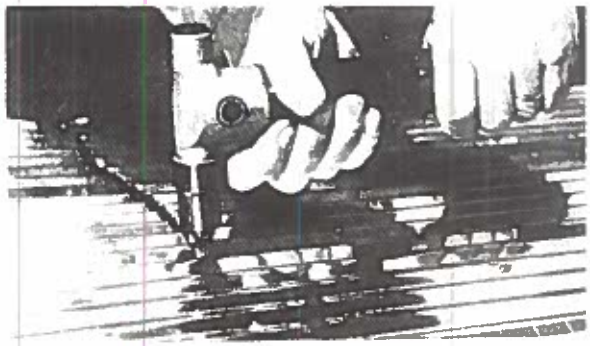
The correct choice of socket is particularly important in the case of an impact wrench, for this is subject to considerable stressing and needs to be stronger than conventional sockets of similar size. Sockets for impact wrenches are normally made of heat treated alloy steel.

For convenience of handling, impact wrenches commonly have a D-shaped handle with, often, a side handle. However, since the torque is imparted as a series of very rapid impact blows torque reaction is comparatively low, as far as the operator handling the tool is concerned.

In smaller sizes, impact wrenches may have D-shaped lever or pistol grip handles.

Impact wrenches may also be used for stud driving, screwdriving in the case of larger woodscrews, machine screws and self-tapping screws, and various other duties, using suitable bits. Stud runners are usually standard forms of nut runners fitted with stud setting attachments.

A torque control ratchet wrench from Gardner-Denver combining high torque repeatability with access to awkward bolt heads.



Nibbler (Atlas Copco)

CP 42 Air Nip pneumatic wire cutter (Consolidated Pneumatic Tool Co).

Shears and Nibblers

Two basic types of action are employed in metal-cutting tools of this type —

(i) a definite shear action provided by two blades working in the manner of a pair of scissors; and

(ii) a nibbling action provided by a cutter punching out small portions of the sheet material as it progresses along the cut line.

Power in either case is normally provided by a vane type motor driving through reduction gearing (usually through a right angled head).

In the case of shears the cutting blades are visible during operation, thus a marked line can readily be followed. With a nibbler the cutting elements are normally obscured by the head, demanding the use of a template for accurate cutting. Both types of tool, however, have the same application and in addition to sheet metal working can be used for cutting other materials such as non-metallic sheets too hard to be sawn.

Typical sizes range from about 3—15 lb (1.4—2.25 kg) in weight, individual tools normally being rated according to their capacity for cutting various gauges of metal. The larger models have a capacity of up to 12 swg (2.6 mm) in sheet steel and may achieve cutting speeds of up to 10 ft/min (3 m/min) in this thickness of mild steel sheet. Nibblers may have a slightly higher capacity, eg up to 8 swg (4 mm) in mild steel sheet.

Tube Expanders

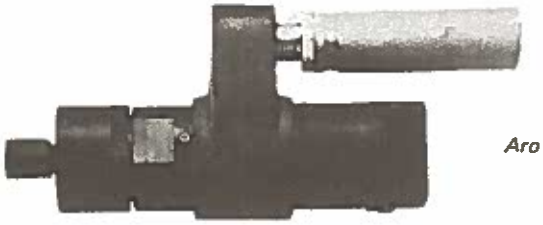
Air drills of 1 in (25 mm) and upwards may also be used as tube expanders and are able to expand up to about 150% of the drill capacity — eg a 1 in (25 mm) capacity air drill may also be rated for tube expanding up to 1½ in (40 mm). Impact wrenches also may be used for tube expanding operations. For heavy duty tube expanding work, however, special tube expanders are normally

used which may employ either a vane type or piston motor. Free speed is usually very low with extra heavy duty tube expanders (eg 20 to 40 rev/min) with stall torques ranging up to 3 000 ft/lb (415 m/kg).

Circular Saws

This type of tool is purely a conventional circular saw driven by an air motor via a spur train or worm reduction gearing (in the case of portable units). It can be considered the equivalent in size and scope of a portable electric saw, although generally lighter and more compact for a given capacity because of the higher power/weight ratio of an air motor.

Motors used are normally of the vane type. Various additional attachments may be fitted for expanding the scope of the saw for grooving, rebating, etc, similar to electric motor powered saw sets. Reciprocating blade saws may also be powered by air motors as jigsaws, power hacksaws, etc, as well as for use as power files. In such cases exhaust air is usually direct to blow over the workpiece and so clear the cutting area of dust and swarf.



Aro power and milling motor.

Tool Chucks

Air-operated chucks offer many advantages over normal manual chucking arrangements, particularly as regards the speed with which workpieces can be mounted on or removed from the machine.

In a typical air-chuck, operation of the chuck is controlled by movement of a piston in an air cylinder, this movement being transferred by toggle linkage to the chuck jaws. Air pressure to the cylinder is normally under the control of a reducing valve enabling the chuck jaw pressure to be regulated as necessary, to avoid distortion or crushing of the workpiece. Further pressure controls may be provided to enable the operator to switch from the high jaw pressure necessary for roughing to a lower jaw pressure for fine finishing, thus relieving the chuck of continual high pressure clamping and consequent strain.

Similar principles of operation are embodied in the pneumatic vice, made in sizes for both workshop and foundry use, pneumatic clamps, *etc*. Again speed of operation is the main advantage offered by such tools. Line pressure is multiplied by the cylinder diameter (*ie* the pressure multiplication factor is equal to the cross sectional area of the piston in in^2, relative to a line pressure in lb/in^2). Additional pressure multiplication can be obtained by inserting a compound toggle mechanism between the cylinder and the vice or clamp. Clamping pressures of up to 15 000 lb/in^2 (1 000 bar) can therefore be developed without resorting to excessive cylinder bore sizes — *ie* overall multiplication factors of up to 150 are readily achievable. With high pressure multiplication however, jaw movement is usually strictly limited and may amount to only a few thousandths of an inch over the initial setting.

Riveting Hammers

Riveting hammers probably represent the largest in numbers of all the percussive tools and range in size from about 2½ lb (1.1 kg) weight for 1/8 in (3 mm) diameter rivets up to 25 lb (11.3 kg) in weight for hot rivets up to 1¼ in (31 mm) or 1½ in (37.5 mm) diameter. They are also designed for specific industries. Thus aircraft or lightweight riveting hammers generally range in piston diameter from 3/8 in (10 mm) for dural rivets of 1/16 in (1.5 mm) to 3/32 in (2.4 mm) diameter up to ½ in (12.5 mm) piston diameter for rivets of 3/16 in – ¼ in (5–6.4 mm) diameter in the softer materials. Slightly heavier models may be used for general metal working duties, and heavy-duty models for the heavier engineering industries, such as shipbuilding. In addition riveting hammers may be adapted for dimpling, scaling, caulking, beading, light shipping, etc.

Riveter performance varies with the particular class or application for which the tool is designed. Thus fast action riveters with short strokes are normally used for working ferrous metal rivets, with speeds up to 5 000 blows per minute. For riveting other metals which would work-harden under repeated hammer blows, slower hitting longer stroke riveters are used. For other work a 'one-shot' riveter may be used, where the tool delivers only a single blow at a time on manipulation of the control lever.

The impact achieved is governed both by the piston diameter and the stroke, the longer the stroke (for a given piston size) the greater the impact. Moderate to long strokes are normally chosen for lightweight riveters, together with spool valves for 'timing'. Construction follows that of chipping hammers, except that the cylinder is usually hardened steel with the bore ground and honed. With larger sizes of riveters strokes may be shorter (in proportion to size), although again there are numerous variants. Some models may incorporate adjustment of force and number of blows per minute to adapt to various work requirements.

Various types of rivet sets can be accommodated, both for driving different rivet heads and for different rivet materials. In addition a 'holder' or 'holder-on' may be incorporated to back up the rivet so that the head is not mutilated and completely fills the rivet cup. Other forms of head may be fitted to adapt riveters as nail and spike drivers, etc, also for pressing, punchwork and stamping. Thus the basic riveter may be made to perform a whole range of duties.

Special types include compression riveters for squeeze-riveting, pop-riveting, eyeletting, etc. Complementary tools include rivet cutters and rivet busters.

It is virtually impossible to give specific figures for air consumption with air riveters since sizes and types vary enormously. As a generalization, however, the heavier tools for hot riveting usually have consumption figures of the order of:-

10–20 ft^3/min (0.3–0.6 m^3/min) for hot rivets up to ½ in (12.5 mm) diameter
25 ft^3/min (0.7 m^3/min) for hot rivets up to ¾ in (19 mm) diameter
25–28 ft^3/min (0.7–0.8 m^3/min) for hot rivets up to 1 in (25 mm) diameter
28–30 ft^3/min (0.8–0.85 m^3/min) for hot rivets up to 1¼ in (31 mm) diameter
35 ft^3/min (1 m^3/min) for hot rivets up to 1.3/8 in (35 mm) diameter
40 ft^3/min (1.15 m^3/min) for hot rivets up to 1½ in (37.5 mm) diameter

One-shot riveters for aircraft type rivets usually have a capacity of 10 to 20 rivets per ft^3/min and rivet busters a consumption of 2 to 5 ft^3/min (0.5 to 0.15 m^3/min) per rivet, depending on size.

Foundry Equipment

PNEUMATIC TOOLS and machines under this heading comprise drop forging hammers, power hammers and forging presses. Cylinder sizes for drop forging hammers may range up to 48 in (1.22 (1.22 m) and strokes from 36–72 in (0.91–1.83 m) with the maximum rate of working ranging from about 200 blows per minute for the smallest sizes down to 60 blows per minute for the largest sizes (although older designs may be much slower working). Forging presses are cylinder-actuated machines and may be all air-operated for small capacities, or powered by air-hydraulic cylinders for higher ratings.

Wide use is also made of a variety of other air tools for forge work, such as scaling hammers, chippers, air-hoists, etc, which are dealt with under the relevant sub-headings.

Foundry Work

A wide variety of pneumatic tools is used for foundry work, embracing both portable tools and static installations such as forging hammers, fettling grips, sand sifters, sand blasters, jolt squeeze machines, etc. Compressed air requirements are, therefore, highly variable from foundry to foundry and demand may range from under one hundred to many thousands of ft^3/min, according to the size of the foundryand chosen tool types. As regards the latter, pneumatic tools are, to a large extent, directly competitive with electric powered tools and actual compressed air demand is largely a matter of the balance arrived at between compressed and electric power.

Some tools are specifically designed for foundry purposes – eg sand rammers. Others, such as grinders, are general purpose workshop tools which find equal application in foundries. Some typical tools and their uses are summarized briefly:-

Sand rammers – are used specifically for compacting moulds and may be either bench type or floor mounted.

Core box vibrators – designed to clamp to a core box (often by an integral pneumatic vice, but alternatively by manual clamping). These are used to loosen moulds so that cores can be withdrawn free of flaws, etc, and also avoid damage to the boxes. Core box vibrators may also be used during the initial stages of making the cores for settling the sand to give a satisfactory core.

Fettling – chipping scaling hammers and grinders are widely used for fettling, with a wide variety of designs and weights available for different duties.

Grinding – for general fettling, high speed wheels are used for rapid stock removal and may run at surface speeds up to 10 000 ft/min. Softer and finer wheels are required for surfacing and finishing, speed depending on wheel diameter and type. All pneumatic grinders can normally take a wide variety of different grinding wheel shapes, as well as wire brushes, etc. The grinders themselves may be straight or angled, choice being largely dictated by the type of work involved and corresponding ease of manipulation of the two types of grinder.

Jolt squeeze machines – used for consolidating sand moulds. The moulding box containing the sand is placed on the machine table. In the case of a shallow mould the sand can be consolidated by squeezing it between the table and an air operated platen ('squeeze'). For deeper moulds, more satisfactory consolidation is achieved by raising the table and then allowing it to drop suddenly ('jolt').

Fettling grips – normally comprise a piston operated clamp which is designed to facilitate removal of fins or flash from hot castings, drop forgings, etc. Fettling grips can accommodate a variety of sizes and shapes of castings, and hold them rigidly against a solid steel plate. Excess material can then be removed by any suitable tool, such as a pneumatic chisel or grinder, with the casting or workpiece securely located all the time.

Sand blasting machines – these are widely used in foundries for cleaning up and finishing castings, forgings, etc. Such machines are simple devices for blowing a concentration of solid abrasive particles (shot, sand, etc), through a nozzle onto the surface of the workpiece at high velocities by means of compressed air.

Percussive Tools

Chipping hammers range in size from about 3–15 lb (1.4–2.3 kg) weight and are normally fast-hitting hammers carrying a chisel blade in place of a hammer head. They are generally grouped in three categories – 'light', 'medium' and 'heavy'. Light chippers are also used for scaling, caulking, beading and similar duties. The heavier tools are used mainly for chipping duties, such as chipping castings, removing flaws in steel billets, etc, but may also be used for riveting. Since the duty can readily be varied by fitting a chisel blade or tool steel of a suitable size and shape to suit different operations, it is difficult to distinguish between smaller chipping hammers and caulking hammers, chisels and scalers, etc, although the latter may be produced as specific types. Typical air consumption figures for chipping hammers are: light 10–15 ft^3/min (0.28–0.42 m^3/min); medium 18–20 ft^3/min (0.5–0.6 m^3/min); heavy 24–28 ft^3/min (0.7–0.8 m^3/min); at 80 lb/in^2 (5.6 bar). Blows per minute range from about 4 000–45 000 for light hammers; to 1 500 –1 650 for heavy hammers. Typical accessories for chipping hammers include various chisels with hexagonal or round shanks.

Light duty chipping hammer (Thor Tools).

Scaling Hammers

Scaling hammers are usually the smallest and lightest of the hammer tools and are commonly produced with valveless construction for high speed of operation of up to 6 000 blows per minute.

In such cases the piston itself may be used as the hammer with the tip or working face hard faced to resist wear. The blows are relatively light, although rapidly delivered, and so brinelling of the surface being treated is usually no problem. Multiple piston types are also produced, either as hand held tools or mounted on a frame for multiple tool work. Other scaling hammers are designed with detachable or interchangeable steels with positive retention for the steel when fitted, and with mechanical valving. There are also various forms of handles and controls favoured by different manufacturers. Most scalers are designed so that exhaust air is directed onto the work surface to assist in cleaning, whilst some may also incorporate a specific blower arrangement rather on the lines of the rock drills.

Scaling hammers – or more correctly, scalers, since not all such tools are fitted with specific hammer heads – are normally made in sizes ranging from about 3–8 lb (1.4–3.6 kg) weight, with strokes ranging from about 9/16–1¼ in (14–30 mm). They are intended primarily for cleaning off paint or scale from large expanses of sheet metalwork, etc, but may also be suitable for other duties. Thus a small scaler, being lighter than a chipping hammer, may be more convenient to use for light metalwork operations, such as peening, beading, etc. Typical air consumption figures for scaling hammers are 5–7 ft^3/min (0.19–0.20 m^3/min) for the smaller valveless types, up to 20–25 ft^3/min (0.6–0.7 m^3/min) for the largest scalers employed for boiler tube work, etc.

Weld flux scalers are simply smaller sizes of chipping hammer intended specifically for the treatment of welds. They may also be used for other scaling duties within their work capacity.

Sand Rammers

The sand rammer is a more or less conventional percussive tool adapted to take a rod attached to the piston to which may be fitted ramming butts or peens (usually of rubber). In some cases the rod is threaded so that extension can be fitted for ramming into deep pipe moulds. Hammering rate is usually of the order of 650 to 800 blows per minute.

Sand rammers are used for foundry work and their design is specific to such use (ie differs from the contractors' rammer). Sizes range from about 4–32 lb (1.8–14.5 kg) in weight, the smaller models being used for bench ramming or light core or mould work and the heavier type for heavy ramming and building up large moulds, etc. Physical sizes range from 5/8 in (16 mm) bore and 2 in (50 mm) stroke up to 1½ in (37.5 mm) bore and 6 in (150 mm) stroke (the latter being equivalent to a back-fill tamper in work capacity). A balanced design is desirable so that the tool 'walks' over the work.

Bench type sand rammers have air consumption figures ranging from about 5 to 15 ft^3/min (0.14 to 0.4 m^3/min) at 80 lb/in^2 (5.6 bar). Floor type (heavier) rammers have a typical consumption of 17–20 ft^3/min (0.48–0.56 m^3/min) in the medium sizes (1 in bore) (25 mm), and 23–30 ft^3/min (0.65–0.85 m^3/min) in the heavy sizes.

Spray Techniques

THE THREE different methods of spray painting are summarized in Table I. All three methods can be used for cold or hot spraying, in manual or automated systems. Choice of system depends on a number of factors such as cost of equipment, cost of paint, size and nature of object to be sprayed, type of finish to be sprayed (see Table II) and quality of finish required. Each system has its advantages and disadvantages. In some cases, a combination of systems may be used — *eg* low pressure spraying for applying primers, followed by hot spray or electrostatic spray for finish coats.

Cold or Hot Spray

Hot spraying allows a paint to be sprayed in more concentrated form, flows better and produces a more homogeneous coating. It also results in better paint economy, whichever system is used.

Hot spraying is particularly suited to the application of thicker coats and is often preferred for priming coats. Low pressure hot spraying is the preferred choice for high quality (gloss) finish coats.

Operating principle for hot spraying is illustrated in Fig 1. Paint is held in a heated container (with either direct or indirect heating), and when spraying is not in progress circulates in a closed

1 — Paint heater
2 — High-pressure pump
3 — Spray gun
4 — Back-pressure regulator
5 — From the paint container or paint circulation system

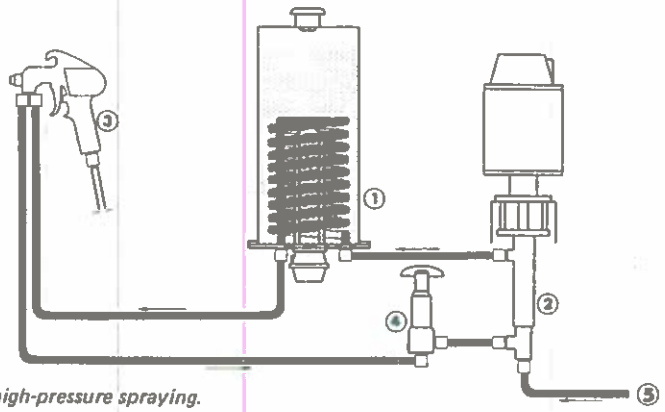


Fig 1 Operating principles for hot high-pressure spraying.

TABLE I – SUMMARY OF SPRAY SYSTEMS

Type	Characteristics	Advantages	Disadvantages	Applications
Low pressure (up to 10 bar)	Compressed air used both for paint transport and atomization	Quick and easy adjustment of fan width and paint quantity. Low cost equipment.	Heavy paint fog and high paint losses. Limited capacity (2 lit/min maximum with pressurized feed).	Car bodies, office machines, refrigerators, furniture, high class work; spraying primers.
High pressure (up to 360 bar)	Necessary pressure generated by a piston pump.	High capacity; thicker coatings; minimal paint mist. Suitable for high viscosity paints.	Higher cost equipment, less control of finish.	Painting large objects, ships, buildings, etc; applying protective plastic coatings.
Electrostatic	Can use either liquid or powder paints.	Particularly suitable for automated systems; wrap-around coating characteristics. Superior paint economy.	High voltage equipment; objects must be earthed; more stringent safety regulations.	Car bodies, steel tube items, cycle frames, fences, etc.

TABLE II – SUITABILITY FOR VARIOUS TYPES OF PAINTS AND FINISHES

Type of Paint or Finish	System(s)	Remarks
Solvent type (drying by solvent evaporation)	Low pressure spray High pressure spray Electrostatic spray	Cold or hot spraying as applicable. (Dispersion-type paints are not sprayed). Chlorinated rubber paints may not be suitable for spraying.
Air drying by oxidation	Brush painting preferred	
Evaporation and chemical reaction type	Low pressure spray High pressure spray Electrostatic spray	Electrostatic spraying limited to special paints.
Stoving enamels	(i) Low pressure spray (ii) High pressure spray	(i) Hot or cold (ii) Hot Electrostatic spraying not generally suitable.
Bituminous paints	Low pressure spray High pressure spray	Hot spraying preferred.
High zinc paints (organic)	Low pressure spray High pressure spray	Zinc-epoxy paints may also be sprayed electrostatically.
High zinc paints (inorganic)	High pressure spray	

circuit from the pump, heater, spray gun and back to the suction side of the pump. A back pressure regulator is used to adjust the paint quantity returned to the main circuit.

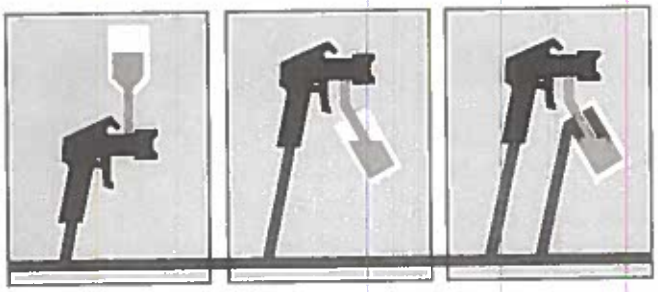


Fig 2 Different paint feed (Atlas Copco)
Left – Gravity feed
Middle – Suction feed
Right – Pressure feed

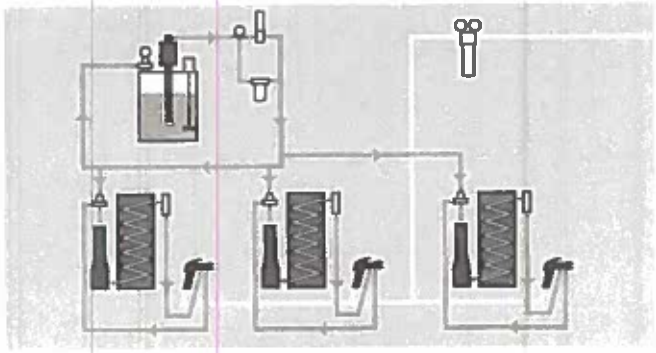


Fig 3 Circulation system for hot spraying.

Paint Transport (Paint Feed)

In low pressure spraying the paint feed may be by gravity, suction feed or pressure feed, the type of spray gun being constructed accordingly – Fig 2. Pressure feed provides a more regular production process and may be used with both cold and hot paint in all systems. A typical circulation system for hot spraying is shown in Fig 3. In a cold spraying system the arrangement is similar except that the heated container(s) are eliminated and the feed is direct from the mixing tank to the spray gun(s).

SECTION 4b

Devices and Machines

Lifts, Hoists and Winches

LIFTING AND hoisting machines can be powered by either air cylinders or air motors. Pneumatic winches are basically conventional winching mechanisms powered by air motors, although some types of winching actions are also performed by air cylinders.

Air motors (or gas cylinders) can be used to advantage where other types of driver are less economic to operate or are excluded because of the conditions involved (*eg* air motors can operate successfully at high temperatures or in explosive atmospheres which could preclude the use of an i/c engine or an electric motor as a driver). An air motor has the further advantage of being very simple to control. Motor hoists are equally suitable for fixed installations or for mounting inside hoist installations running on an elevated I-section rail.

One of the range of Gardner-Denver axial piston motor powered air hoists.

Cylinder Hoists

Cylinder hoists have a more limited field of application than motor hoists for working at extreme or variable heights. Also cylinders tend to be bulky by comparison with air motors of the same lifting capacity. The usefulness of cylinders can, however, be extended by compounding the mechanism, using two or more cylinders.

Another basic difference between the two types is that cylinder hoists are best suited to fixed installations (although many are trolley-mounted). They lend themselves particularly to siting

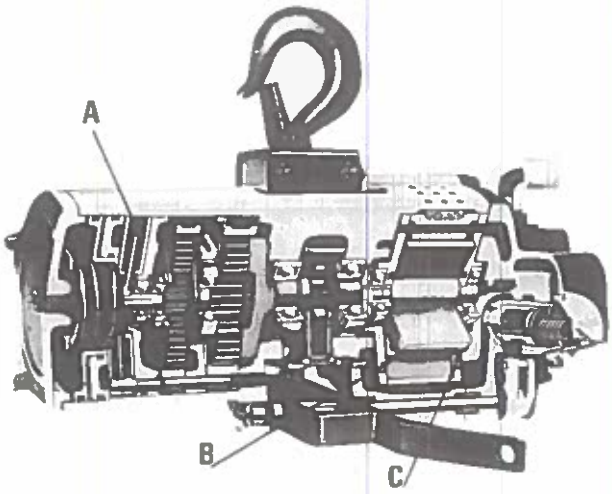
alongside or on the device to be operated, such as a pneumatic loader. For specific lifting applications they have the advantage over air motors that they can provide greater load stability and a more rigid system.

Air Motor Hoists

The specific advantages normally provided by motor-powered air hoists include:

(i) small, lightweight compact construction, which can be allied to general solidness.
(ii) low maintenance requirements.
(iii) variable lift speed.
(iv) safe (*ie* undamaged) if stalled or overloaded.
(v) high working cycle (as much as 100% if necessary).

The characteristics of a motor hoist or winch are determined by the type of motor fitted. Thus a piston motor would be used where high starting torque is required. Heavy duty or general purpose air hoists are usually fitted with piston motors. Vane motors are, however, also widely used for air hoists and winches, particularly where a compact lightweight unit is required.



Thor rotary hoist. A — disc brake. B — unitized motor unit. C — vane motor.

Hook mounted hoists with pendant control. (Thor Tools).

The basic hoist unit can be equipped in different ways for lifting — *eg* with roller or link chain, rope or pendant control. Rail-mounted units can also incorporate traction drive to provide traverse motions both along a horizontal rail and up gradients. An ingenious extension of this is the mounting of the hoist on a flexible wire rope suspended from a safety booth above the working area. The hoist then runs up and down the wire rope which can also be swung to one side to provide pick-up over a wide area. This form of suspension is limited to smaller capacity hoists — *ie* with a typical lifting capacity of up to 100 kg (220 lb). The general range of compressed air (motor driven) hoists is from 10—10 000 kg (220 lb—10 tons). The all-up weight of an air hoist can be as low as 1.5% of its lifting capacity (in larger sizes).

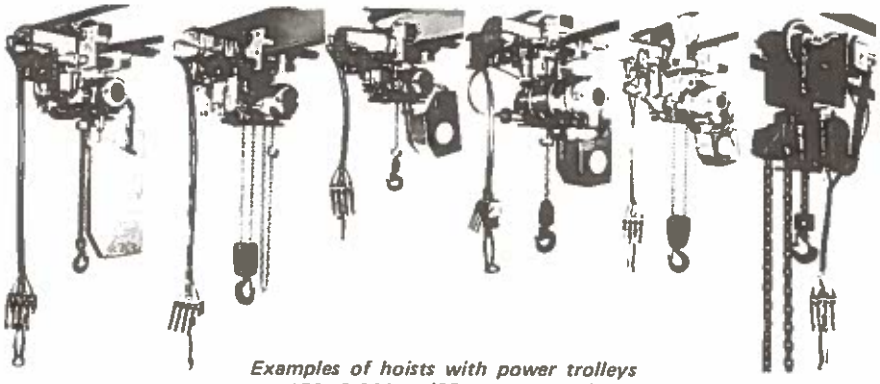
Lifting speed is largely governed by the reduction gear ratio employed in the design of the hoisting unit. The actual speed achieved is also dependent on load. Typically the full load speed is 50% of the free (unloaded) speed. Some representative performance figures are given in Tables IA and IB.

TABLE IA – TYPICAL PERFORMANCE OF AIR HOISTS

CAPACITY		LIFTING SPEED			
		Free		Full Load	
kg	lb	m/min	ft/min	m/min	ft/min
250	550	50.0	165	27.0	88
500	1 100	25.0	82	14.0	46
1 000	2 200	12.0	40	7.0	23
2 000	4 400	6.0	20	4.0	13
3 000	6 600	4.5	15	3.0	10
6 000	13 200	2.2	7	1.5	55
10 000	22 000	1.9	6	1.0	3

TABLE IB – TYPICAL PERFORMANCE OF AIR HOISTS

Capacity	AIR CONSUMPTION			
	Free		Under Load	
	m^3/min	ft^3/min	m^3/min	ft^3/min
250	1.7	60	1.60	56
500	1.7	60	1.60	56
1 000	1.7	60	1.60	56
2 000	1.4	50	1.15	40
3 000	2.4	85	2.20	78
6 000	2.4	85	2.20	78
10 000	2.4	85	2.20	78



Examples of hoists with power trolleys 150–2 000 kg (330–22 000 lbs). (Deutsche Gardner-Denver).

Air Starters

AIR MOTORS offer advantages over electric motors for starting large capacity i/c engines and gas turbines, where a compressed air supply is also available. Air motor weight is only about 50% of that of an electric motor for the same power and an even more substantial reduction in battery weight is possible. Initial cost is also lower, with substantial savings in depreciation and maintenance. A further advantage is that starter operation is independent of battery condition. The air starter is powered from an air tank, either charged from the existing compressor (on vehicles fitted with air brakes, for example) or from a separate engine-driven compressor.

Vane-type air motors are commonly used for starter duties, coupled to a conventional Bendix drive via reduction gearing. For higher powers, impulse turbine motors are preferred, driving through reduction gearing to an inertial starter drive. Turbine motors have the advantage of requiring no lubrication of the drive air (necessary with vane motor starters); and have no internal rubbing surfaces and can thus tolerate prolonged cranking cycles.

Operating air pressure for air starter motors is usually of the order of 7 bar (100 lb/in^2) which is readily available from the air brake system compressor charging the separate air starting tank *via* a one-way check valve. A safety valve is also incorporated, so that the air tank can be kept permanently topped up at full charge when the engine is running, the check valve preventing air bleeding back into the brake system.

Turbine air starter motor.
(Tech Development Inc).

The size of tank needed is somewhat arbitrary, but is normally based on sufficient reservoir capacity to give some five or six starts from a full charge. Some weight saving is possible by making the tank from aluminium rather than steel, although this will increase the initial cost.

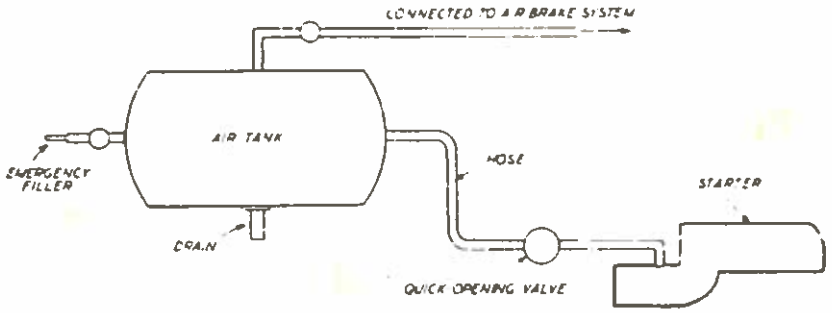


Fig 1

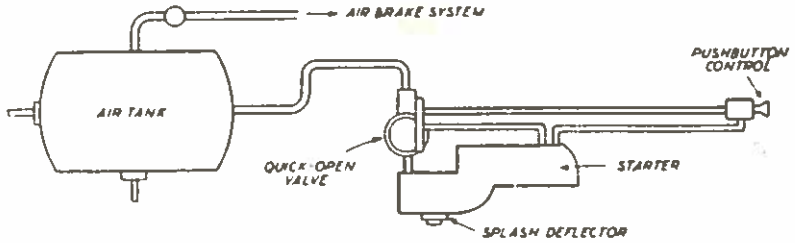


Fig 2

A typical starter circuit is shown in Fig 1. The starter control is a quick-opening valve, normally manually operated, both for simplicity and to divorce the starter system entirely from the electrics. It could, however, be solenoid operated *via* a push-button if preferred, or pneumatically operated *via* a remote control position as in Fig 2. Besides the two connection points, the tank is fitted with a drain to enable condensation to be drained off at regular intervals, and a second valve for emergency filling. This enables the tank to be charged from an independent source if necessary — *eg* a service station air supply, another vehicle's air brake system, or even a gas bottle.

With a stationary system the tank can be eliminated entirely, the motor being connected directly to an existing air supply. Similarly, no separate electrical system is needed in the case of a spark-ignition engine, as magneto ignition can be employed.

Air and Electrics Compared

The complete elimination of electrics on diesel and spark-ignition driver installations can be particularly advantageous in certain applications — *eg* for the savings possible in initial costs, installation weight, reduced maintenance, and so on. If auxiliary electric power is still required, this can be provided by an engine driven generator or alternator, with or without a battery as necessary. If a battery still has to be used, then air starting renders the system independent of battery condition and all services are always available as long as there is sufficient compressed air in the tank for starting — *ie* even with a flat battery, all battery powered services are restored as soon as the driver is started by the air motor.

Air motors are known for their reliability, and their consistency of performance is generally superior to that of electric motors since they are less affected by surrounding conditions. Thus the air motor is dependent only on the availability of a sufficient quantity of air to generate the speed required, and at a suitable pressure to generate the torque required. This is largely independent of the ambient temperature and thus the air motor retains good power characteristics over a very wide range of temperatures. Electric motors, on the other hand, can suffer from a loss of battery power as high as 50% in extremes of low temperature. A further advantage of an air motor is that it can be stalled under full power without damage, and does not place an additional demand on the air supply in a stalled or near-stalled condition. By comparison, a heavily loaded d.c. electric motor will draw increasing current with increasing load, *eg* when starting a very cold engine, and if it is stalled there is the very real risk of burning out the armature, or at least flattening the battery.

Standby Use

Because they are independent of battery performance and battery condition, air starters are attractive for emergency and standby sets. These can be independent units with their own pre-charged air tanks, or can draw their air from a normal air line. The system is flexible enough to adapt to a variety of situations. Thus, with a precharged starter unit driving a portable compressor, the air tank can be recharged as necessary from gas bottles, or even from a manual pump.

Portable Power Units

Air starters are finding increasing application in the field of mobile and portable engine powered units, and even on portable and emergency electric generating sets, to remove their reliance on battery condition. In the case of standby units automatically controlled by line loads, the air valve controlling the air starter can be of solenoid-operated type.

The possibility of operating the air starter on a medium other than compressed air may also exist, *eg* any high pressure gas that may be available. An example here is the use of high pressure natural gas to operate the air starters on gas turbines installed to pump that particular gas. There is no reason why such a 'free' source of compressed air should not be utilized, providing the pressure available is adequate for operating the air motor and the use of the gas in this manner does not represent any hazard. If the available gas supply pressure exceeds the rated limit for the starter, then it is a simple matter to drop this to a suitable level by interposing a reducing valve in the supply line.

Air Springs

THE ELASTIC nature of air contained in a closed chamber and subject to compression provides the principle of the pneumatic spring. Their particular advantage is that they are clean to produce, simple to install, and can be designed to cover a wide range of performance requirements in two distinct fields of application, *viz* –

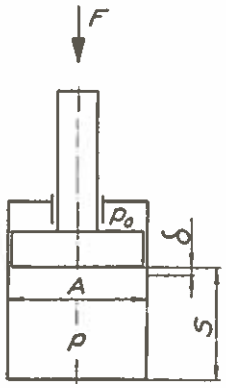
(i) Alternatives to conventional (mechanical) springs for suspensions, *etc.*

(ii) Vibration and shock isolation and control.

The simplest form of air spring is a closed pneumatic cylinder (Fig 1), where the spring force is given by

$$F = A P_1 \left(\frac{S}{S-\delta}\right)^{1.4} - P_O$$

where A = area of the piston
P_1 = initial pressure in cylinder
P_O = atmospheric or ambient pressure
S = stroke
δ = deflection or travel of the piston under force F



E = force
P = internal pressure
P_o = atmospheric pressure
δ = deflection
S = stroke
A = piston area

Fig 1

The spring force is thus an exponential function, instead of the simple relationship of a mechanical spring, *ie* force directly proportional to deflection.

Spring rate is determined as a function of the volume/area change and is widely variable by altering the geometry. It is thus possible to design pneumatic springs with a wide range of dynamic characteristics. The main limitation of the simple piston-and-cylinder form of air spring is reliance on a piston seal and the sliding friction involved. This can be eliminated by adopting a flexible bellows form providing positive sealing with no break-out or sliding friction.

The working characteristics are dependent on the volume ratio between the spring in its initial state and compressed state and corresponding pressure change, *viz*

$$P_2 = P_1 \left(\frac{V_1}{V_2}\right)^{\alpha}$$

where P_1 = initial air pressure inside spring
P_2 = final pressure inside spring
α = is an exponential, typical value 1.38

Operating pressure (P) is determined by the load (L) and the effective area (A) of the air spring. In turn, the effective area is determined by the effective diameter (D_e) of the spring. This will vary with the form of the bellows – *eg* see Fig 2. The operating pressure in terms of measurable parameters then becomes:

$$P = \frac{L}{0.7854\ (D_e)^2}$$

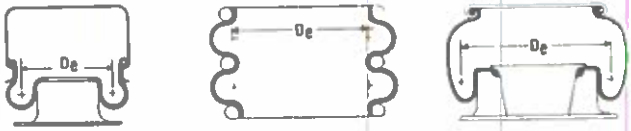


Fig 2

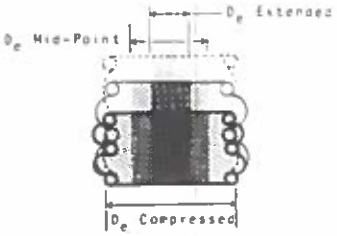


Fig 3

As the spring is compressed by increasing load, its volume V_1 decreases to V_2; and at the same time its internal pressure increases from P_1 to P_2 (both pressures being absolute). Related to the formula given, this means that in plotting a dynamic load deflection curve it is only necessary to multiply the new (measured) *gauge* pressure by the effective area to determine the new spring load after a volume change due to spring movement. However in many configurations the effective area also changes as the spring moves (*eg* see Fig 3). Thus the instantaneous effective diameter at intermediate positions must be used in calculating the new load after spring movement.

The rate of a typical air spring of this type will increase as the static load increases. This means that the equivalent static deflection and rational frequency remain essentially unchanged. This is a particular advantage where an air spring is used to support a varying load, *eg* as in a vehicle suspension system.

Configurations

The most common configuration of air spring is a bellows form with one or more convolutions. Some form of external control may be necessary with more than one convolution, and this can be provided by wire rings or a wire spiral. Alternative forms of suitable bellows are rolling sleeve and rolling diaphragm types.

The usual choice of material for the bellows is an elastomer reinforced, as necessary, to withstand the likely pressure changes handled by the chamber. Elastomeric bellows, generally, offer simpler fabrication and freedom from fatigue, as well as excellent resistance to corrosion, *etc*. The material most commonly used is nylon reinforced rubber.

With elastomeric bellows the maximum static air pressure in the spring is normally restricted to about 7 bar. Actual working pressure range within the chamber would then be of the order of 0.7–7 bar. Higher maximum pressures of up to 12.5 bar can be accommodated by nylon-reinforced neoprene bellows. If working at higher air pressures is required, then metal bellows would normally be used.

The static load range for a typical single-convoluted bellows with a maximum design working pressure of 7 bar could extend from 227–25 000 kg depending on physical size. Springs of this type are particularly suitable for anti-vibration or shock-insulation mounts.

Where greater deflections have to be accommodated, the stroke range can be increased by increasing the number of convolutions. Very roughly, the stroke is increased in direct proportion to the number of convolutions. Thus Fig 4 shows a two-convolution pneumatic spring with approximately double the stroke of a single-convolution spring of the same diameter, and Fig 5 a three-convolution pneumatic spring. In this case the stroke is about 2.7 times that offered by a single-convolution pneumatic spring of the same diameter.

Fig 4

Fig 5

'Pneuride' bellows have a wide range of applications both as linear actuators and vibration isolation devices, are available with nominal diameters ranging from 6 to 16 ins and either 2 or 3 convolutions to provide a variety of working movements. Moulded in flexible reinforced neoprene rubber, the units can be operated on air pressure up to 8 bar (120 lb/in^2). Mounting is by means of 4 M.8 metric threaded studs in the steel end plates and there is a single ½ in BSP inlet port.

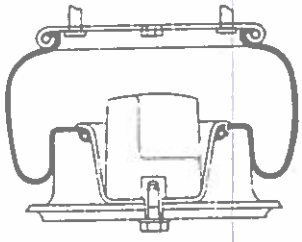


Fig 6

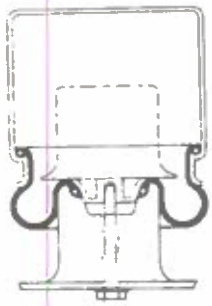


Fig 7

Pneumatic springs, particularly those of the multi-convolution type and rolling sleeve type (Fig 6) are suitable for direct application as vehicle springs in conjunction with such damping devices as may be necessary. Springs of these two types can generally be used on trailers without damping.

Single convolution bellows and rolling diaphragm configurations (Fig 7) produce stiffer springs which are more suitable for shock and vibration absorption.

Load capacity of a pneumatic spring can be as high as 500–1000 kg per kg weight of spring, depending on the design size and type. This figure compares favourable with most other types of conventional spring. The natural frequency can also readily be selected to come within prescribed limits by suitable attention of geometry. This normally falls within the range of 20 to 150 cycles per minute, or appreciably lower than that of most other spring elements. The natural frequency is a constant factor with a pneumatic spring, since any given design produces essentially the same static deflection or natural frequency under all static load conditions.

The performance of pneumatic springs can also be modified by connecting the (normally) sealed cylinder to a suitable external circuit, *eg* see Figs 8 and 9.

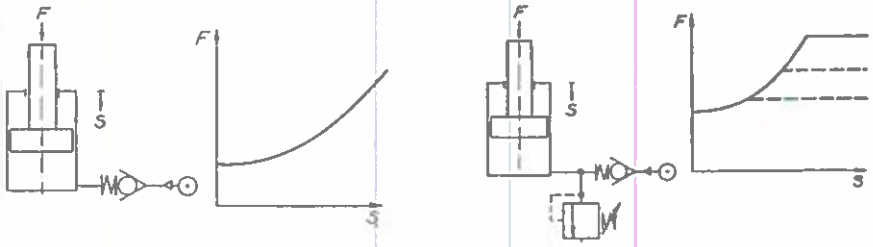


Fig 8 Spring force diagrams — cylinder with non-return valve (left) and with relief valve limiting spring force (right).

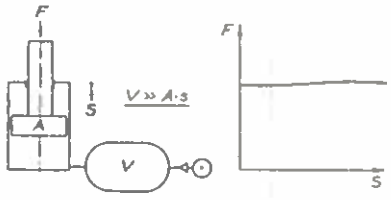


Fig 9 Force diagram with air spring connected to a large reservoir.

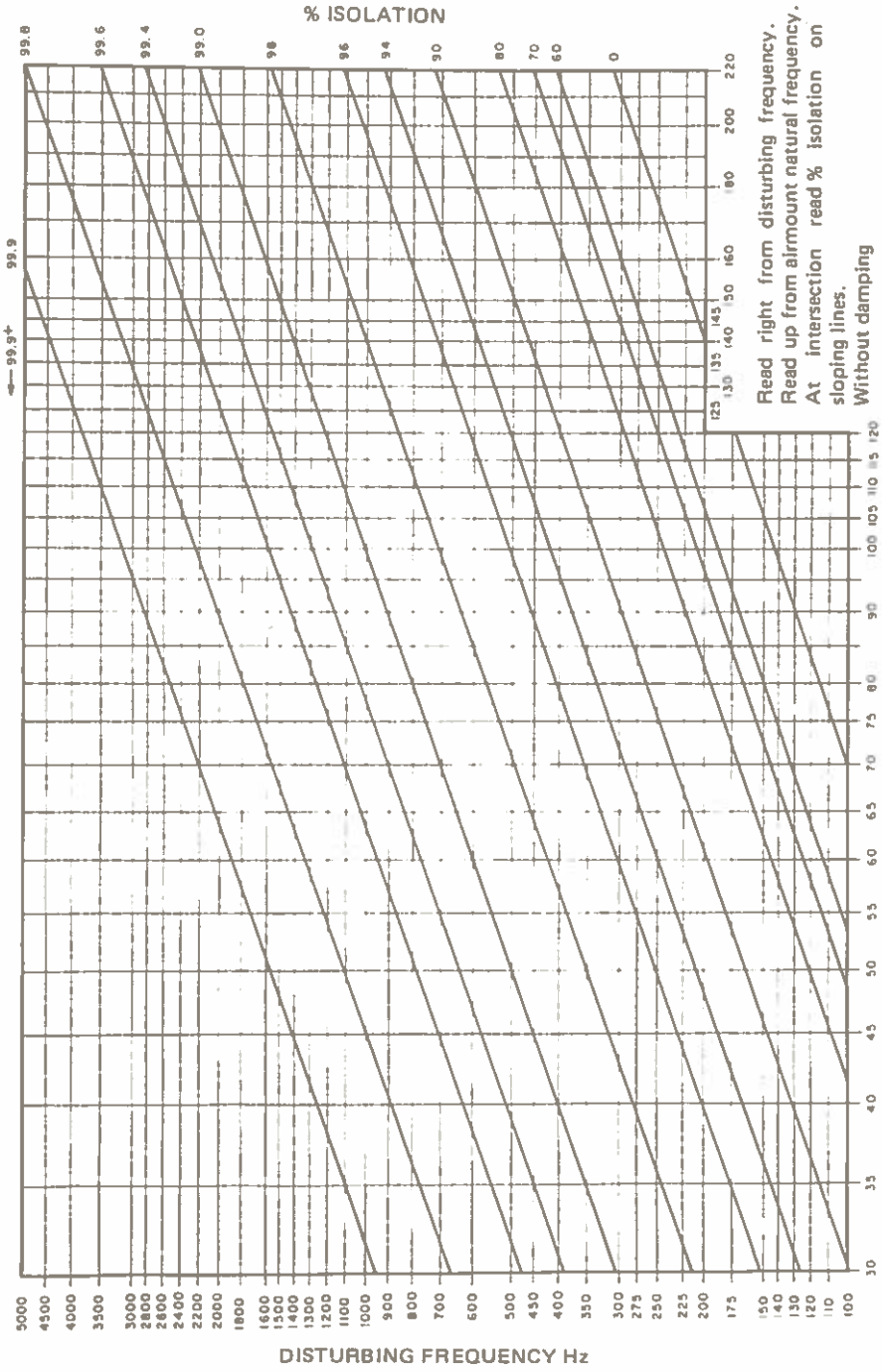
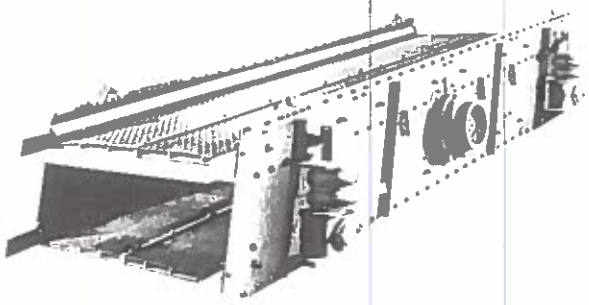


Fig 10 Vibration Isolation Chart for Firestone Airmounts



Firestone Airmount isolators, in double convolutions, are shown here in a typical application. Steel girdle rings increase stability.

Air Springs as Isolators

Natural frequencies obtainable with typical designs of air springs range from about 5 Hz to 1 Hz. Connected to an auxiliary air volume it may be possible to extend this range down to about 0.5 Hz (*eg* see Fig 10). Thus excellent isolation of disturbing frequencies down to as low as 10 Hz should be possible with suitable design.

Damping will vary from a damping ratio of 0.01 for convoluted air springs to 0.1 for rolling diaphragm types. Control of damping achieved can also be obtained with various forms of orifice damping.

Convoluted isolators have a lower stiffness rate laterally than vertically, lateral rate increasing as the air pressure and operating height increases. The ratio of lateral rate of vertical rate is typically of the order of 0.2 to 0.3.

Rolling diaphragm isolators can be designed to have a lateral rate equal to, or greater than, the vertical rate, the rate ratio varying with the configuration of the metal surfaces.

Rolling slave isolators are unstable laterally. They are therefore only used where the vibratory mass carried is restrained against horizontal movement by some other means.

With all types of air (spring) isolators stability requirements must be considered in some detail because of the inherent softness of the springs. These are not easily met when the centre of gravity of the supported mass is low.

Table I is a general selective guide to air springs for different applications.

TABLE I – SELECTION GUIDE TO PNEUMATIC SPRINGS

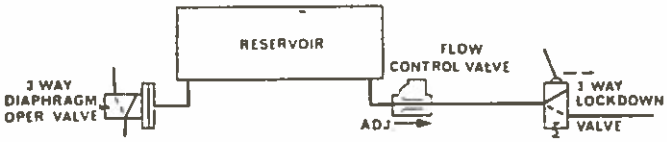
Requirement	Type
Maximum load	Bellows
Maximum load and minimum deflection	Single-convolution bellows
Maximum load and maximum deflection	Three-convolution bellows
Maximum stroke	Rolling sleeve or three-convolution bellows
Minimum stroke	Any type
Minimum axial space	Single-convolution bellows
Minimum lateral space	Rolling sleeve
Lowest natural frequency	Rolling diaphragm
Highest natural frequency	Single-convolution bellows
Greatest range of natural frequency	Rolling diaphragm

Pneumatic Timers

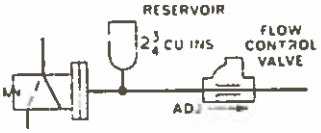
TIMERS OPERATING on pneumatic principles can be of the following types:

(i) Time-delay circuits.
(ii) Piston/air bleed.
(iii) Timing valves.
(iv) Air motor clock drives.

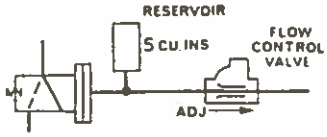
Diaphragm valves in conjunction with flow control valves are used to produce time-delay circuits. Three examples of basic circuits of this type are shown in Fig 1.



Basic time-delay circuit.



Delay-time up to 12 seconds (at 80lb/in^2).



Delay-time up to 16 seconds (at 80 lb/in^2).

Fig 1

Piston/air bleed timers may be of two types. The simplest is a spring-loaded cylinder with an adjustable air bleed at the blind end – Fig 2. The performance of such a simple unit has distinct limitations. Positive timing is dependent on a perfect piston seal on the closing stroke and positive adjustment of the air bleed. Any setting of bleed rate timing (actual time of stroke) will tend to be variable with air density and temperature. Nevertheless, quite consistent results can be achieved from such timers if they are well designed and made, particularly in the small sizes. The attraction

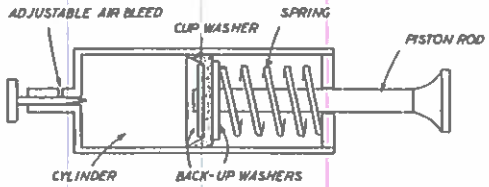


Fig 2

is that they can be made extremely small and light — *eg* the cylinder may be as small as ¼ in diameter and the total weight of the timer ¼ oz or less. They are not, however, generally suitable for precision timing.

Substantially higher output forces can be achieved by replacing the air bleed by a fixed or variable orifice, eliminating the spring and working the timer in the reverse mode. Timing is controlled by the orifice and the output force available is equal to the product of the piston area and the difference between the supply pressure and pressure drop through the orifice. This type of timer, of course, requires a compressed air supply, whereas the simple air bleed type does not.

To produce longer time delays and a rotary output movement the piston can be used to operate a ratchet. The air supply is arranged so that, when the ratchet is operated, the air supply is removed from the piston allowing it to return by means of an integral spring. In such a system the air flow to the piston controls the speed at which the ratchet rotates. Such a stepping mechanism can be used to provide longer time delays or to operate any rotating equipment.

A more refined method of producing time delays is by the use of a mechanical escapement timer. It is possible to wind up the spring using an air-operated piston and when the escapement timer has completed its timing, it can release a lever which will mechanically operate a pneumatic valve. The output from this valve will produce the timing signal at the end of the time delay set on the escapement timer.

In general, piston/air bleed timers tend to be suitable only for relatively short timing periods (*eg* up to 1 minute), with their accuracy largely dependent on ambient temperature. They are also rather prone to malfunction due to their need for free movement of the piston and for clean air to prevent the orifice becoming clogged. In spite of these drawbacks this type of timing equipment is quite commonly used in applications where cost is important but accuracy is not paramount. The timers are generally available with a delay function, *ie* timing is initiated by connection of the air supply and the output is made available after the time delay has elapsed; or with interval function, *ie* the output is made available at the same instant as the air input supply is turned on and is then removed after the time delay has elapsed. There are a number of other variations on these timing duties which can be achieved with this type of equipment.

Various forms of camshaft timers driven by a pneumatic piston/ratchet drive system are available. Depending upon the gear ratios utilized, they can provide cam-shaft speeds as slow or as fast as is required. This type of drive system, however, suffers from poor accuracy and from a fairly high ambient level.

Timing Valves

Timers which work from a definite compressed air supply rather than by a simple piston are more generally referred to as timing valves although similar principles are involved — *eg* a basic time-delay volume is incorporated with timing rate or timing action controlled by an air bleed. Frequently the air bleed path is thread clearance to provide a convenient and practical method of adjustment.

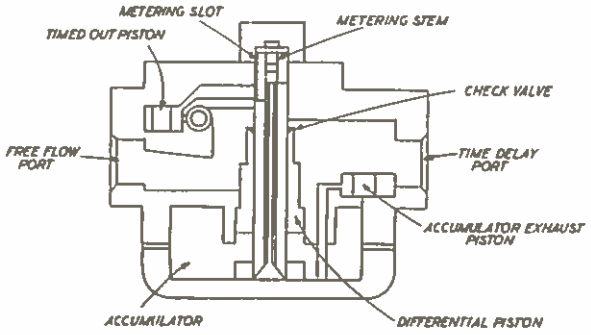


Fig 3

A design of timing valve is shown in Fig 3 where the timing volume is metered by a sharp edged orifice produced by a slot cut through the side of a hollow stem in which a close fitting spindle travels. Flow to the time delay volume is adjusted by altering the position of the spindle along the length of the slot, the time required to change the volume and actuate the valve being determined by the length of slot left open by the spindle.

A particular advantage offered by this system is the infinitely variable adjustment of orifice area possible giving fine resolution, whilst the flow characteristics are those of a single sharp-edged orifice rather than the more usual thread leakage. This gives high repeatability of the order of ±2% at fixed supply pressure and ±5% with a 10 lb/in^2 variation in supply pressure. These figures refer to a specific design, as accuracy of working and repeatability will depend very largely on actual construction and working conditions. For long-term accuracy it would be necessary to ensure that the air supply was clean and free from moisture.

In the particular design shown, three snap-action valves are incorporated in the valve envelope. The main valve is a check valve which controls the flow of working fluid actuated by a differential area piston exposed to the pressure in the timing volume or accumulator chamber. It can be arranged to be either normally closed or normally open, giving either timed-in or timed-out control of flow respectively.

The other two valves are differential area pistons operated by pressure in the inlet and outlet ports. These pistons control flow into and out of the timing circuit.

Pneumatic Sensors

THE MOST common form of pneumatic sensor is probably the interruptible jet, the principle of which is shown in Fig 1. Supply and receiver pipes are aligned axially, separated by a gap. The intrusion of any solid object into the gap, interrupting the jet, causes the pressure in the receiver to fall to atmospheric pressure. This change in pressure is used to operate a switching element controlling an appropriate circuit (*eg* a counter circuit). The switching element is normally a transducer giving an electric signal output.

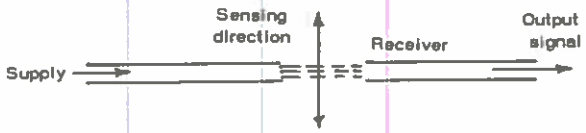


Fig 1

This type of sensor has the advantage that it is not critical on gap dimension, can change its state rapidly (*ie* count rapidly) and is not sensitive to object shape or texture. Its main disadvantage is that atmospheric air is entrained in the receiver, and if contaminated can interfere with its performance.

This limitation can be overcome by lightly pressurizing the receiver in the opposite direction — Fig 2. The main jet then impinges on this secondary jet, applying 'back pressure' which appreciably increases the pressure in the receiver. An object interfering with the main jet cuts off this back pressure, when the receiver pressure falls to its normal pressure level. Under both conditions there is always outflow from the receiver, which thus cannot entrain ambient air.

Both of the above require small nozzle sizes in the supply (commonly provided by a restrictor) to achieve laminar flow. The effective gap length is then mainly the length over which laminar flow is maintained.

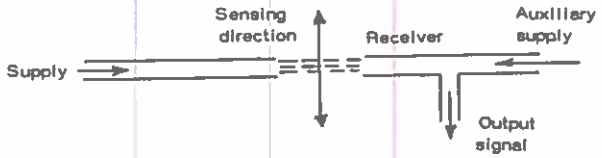


Fig 2

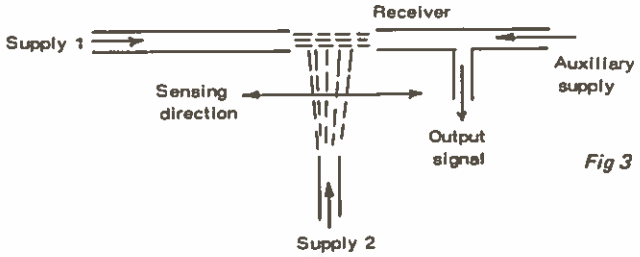


Fig 3

An alternative, shown in Fig 3, uses a third jet placed at right angles to the main (supply) jet. The receiver may or may not be pressurized. This works on the basis of a turbulence amplifier with the main jet flow (normally laminar) being rendered turbulent by impingement of the side jet, but reverting to laminar flow when the side jet is interrupted. The practical measuring gap – which is between the side jet and the main jet – can be made much larger for the same pressure difference created in the receiver.

A more practical form of interrupted jet sensor with backflow from the receiver is shown in Fig 4, backflow being obtained from the same supply, but with pressure reduced by a restrictor. Sensors of this type can be expected to have a maximum gap of about 20 mm working off a supply pressure of the order of 0.1 bar; but larger with higher supply pressures and larger nozzle sizes. The system shown in Fig 5 is capable of working with larger gaps at the same pressure levels. Here a collector is incorporated to supply a second external gap (the sensing gap), with output signal derived from the same position as before. Specifically, this device is known as an *airstream detector.*

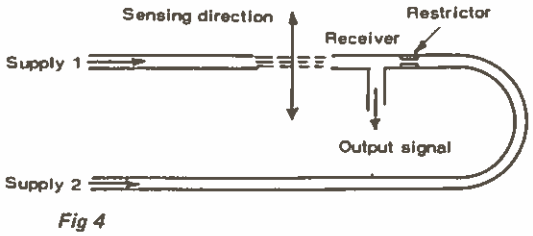


Fig 4

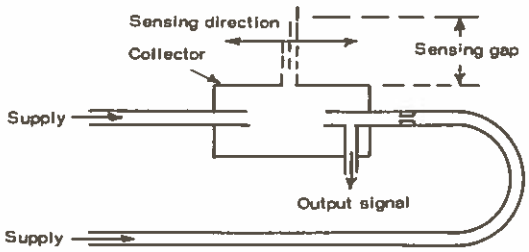


Fig 5

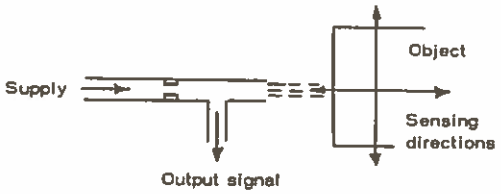


Fig 6

Back-pressure sensors are an earlier type, but are only effective when the object to be sensed can pass quite close to the jet. A single jet is used in this case, the presence of an object modulating the flow and causing a marked pressure change at the output – Fig 6. It can, however, be used to detect objects moving towards or away from it, as well as across the jet path, with suitable signal amplification.

The *conical jet sensor* is a further type where the jet emerges from an annular nozzle and is thus of divergent conical form. Air inside the cone is entrained, resulting in a region of reduced pressure, sensed by the output. The degree of depression, and thus the output signal strength, is modified by any asymmetric distortion of the conical jet resulting from its impingement on a downstream object. It is, however, rather slower in response that other types of interrupted jet sensors and requires more power (higher air flows).

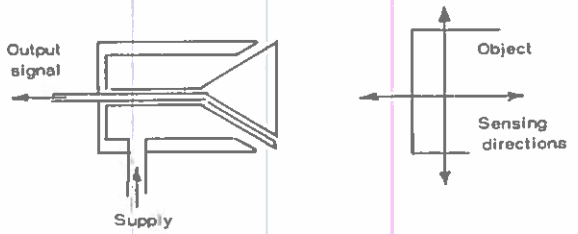


Fig 7

The reverse configuration or cone jet – Fig 7 – employs an annular nozzle to produce a convergent conical jet. When this impinges on an object pressure is increased within the cone, creating a back pressure through the outlet.

All single jet devices are, of course, free from contamination by dirty ambient air. Some, however – and particularly the cone jet – work with particularly small nozzle sizes and so require a supply of clean, uncontaminated air.

Compressed-Air Aspirators

THE COMPRESSED-AIR aspirator can be described as the pneumatic equivalent of the industrial vacuum cleaner, the difference being that the suction force is generated from a compressed air supply rather than a force driven by an electric motor. In a typical unit suction forces down to 0.6 atmosphere can be generated, capable of removing all kinds of liquids and sludges as well as solids up to a specific gravity of 5.0 or greater, through hose sizes up to 3 inches (75 mm) diameter. Compressed-air aspirators can thus solve many cleaning and recovery problems in industry, whenever a compressed air supply is available. A barrelling device may also be fitted to allow any liquids which run out of their containers to be disposed of into normal drains directly.

DEBUS compressed-air aspirator (left) and liquid aspirator (right).

The size of aspirator which can be used, and the suction pipe diameter, depends on the available air capacity. Some typical data are given in Table I.

The length of suction hose which can be used depends largely on the weight of material to be aspirated, when lengths up to 25 metres are often possible with lighter solids, *etc.*

TABLE I – AIR CONSUMPTION OF A TYPICAL RANGE OF COMPRESSED-AIR ASPIRATORS WITH 7 BAR COMPRESSED AIR SUPPLY

Air consumption m^3/min	Suction head mm WG	Intake air m^3/min	Suction pipe by inch	Compressed-air pipe by inch
0.3	2000	2.0	1¼	3/8
0.7	2800	3.0	1¼	3/8
1.3	3400	4.5	2	1/2
2.0	4200	6.5	2	3/4
3.1	5500	8.0	2½	1
4.2	6000	9.5	3	1

Examples of applications are:- Cleaning of machine tools, removal of run-out cooling emulsion, removal of run-out hydraulic oil, aspiration of mercury, oil change at large machines, cleaning of crane runways, aspiration of textile fluffs, sucking out of moulds, draining out of grease separators in food plants, cleaning of tanks and use for industrial fire brigades.

Since a compressed-air aspirator is non-sparking it can also be used in areas with an explosion hazard.

Turboexpanders

TURBOEXPANDERS HAVE a long history of use in air separation, but more recently have come to the fore in the liquid gas industry for the recovery of liquefiable hydrocarbons. They are, in fact, now mainly used for ethane, propane and ethylene recovery and the processing of carbon dioxide, as well as various air separation processes.

Basically, a turboexpander is a high efficiency turbine with numerous special features which make it conveniently usable and reliable for small volumetric flows at low temperature (and often at rather high pressure) usually found in these applications. They have been used in air separation processes for over 30 years.

A turboexpander also generates the deep low temperature refrigeration industrially used for gas separation and liquefaction, and related purposes. It does so by the mechanism of constant entropy expansion, together with the production of power (a by-product). The power comes from the decrease in enthalpy of the stream itself.

The modern turboexpander is a radial reaction turbine, which is a further improvement on the general types of turbine – see Figs 1, 2 and 3. In the common impulse turbine (Fig 1) all of the pressure energy is converted to velocity in the nozzle, and this resulting high velocity stream impinges on U-shaped blades in the rotor. The rotor blades move at half the velocity of the gas jet, and the gas comes out of the rotor blades directed backward with respect to the rotor, to leave without significant residual absolute rotational velocity. This design loses some 8–10% of the available energy in making the U-turn between the rotor blades.

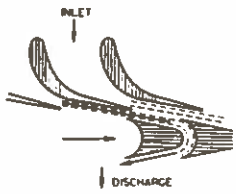


Fig 1 The common impulse turbine.

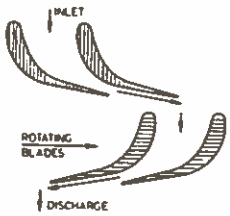


Fig 2 The reaction turbine.

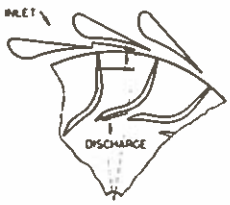
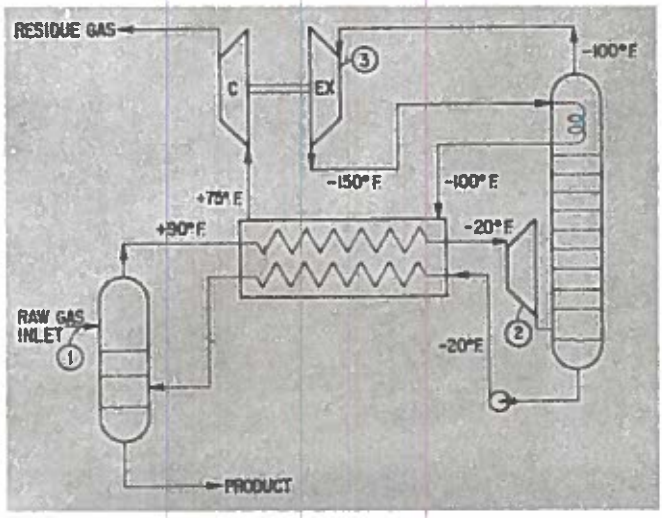


Fig 3 The radial reaction turbine.



Typical turboexpander process for recovering ethane and LPG from natural gas.

The reaction turbine, (Fig 2) is more efficient. In its primary (stationary) nozzles, only half the pressure energy of the gas stream is converted to velocity, and the rotor with a blade speed matching the full jetted stream velocity receives the latter. In the rotor blades the other half of the pressure energy is used to jet the gas out of the rotor backward to exhaust.

Half the pressure drop is across the rotor so there must be a seal around the periphery of the rotor to contain it. Also, this pressure difference across the rotor acts on the full rotor area and creates a large thrust load on the shaft.

A further improvement in turbine design led to the radial reaction type (Fig 3). Compared with the reaction type, it has a reduced discharge diameter. In this design the gas, again half expanded in the primary nozzles and jetted tangentially into the rotor, matches the peripheral speed of the rotor and flows radially inward within the rotor, leaving it at a lesser diameter. This arrangement reduces the velocity required from the secondary (rotor) nozzles with correspondingly lower nozzle friction loss. It also reduces the diameter of the rotor seal. This seal diameter reduction reduces both the seal leakage and also the shaft thrust load. The radial reaction type is the most efficient of the three. It is used in large water turbines, but is not used in large steam turbines because of their large volumetric flow, and because of its inconvenient adaptability to multi-staging.

The radial reaction design is very well suited to turboexpanders for the above reasons. Successful commercial expander processes are dependent upon the design and production of suitable high speed turbine rotors and nozzles capable of reliable operation under extreme conditions of low temperatures and a wide range of pressures.

Air Blowers

THE DESCRIPTION air blower (or blower) is applied to types of air pumps designed to deliver small volumes of air at low to moderate pressure. Some types can also work in the reverse mode, *ie* as exhausters, the Roots' pump being a typical example. Other types have fan-type impellers capable of moving larger volumes of air at very low pressures and are more correctly called *fan-blowers* or turbo-machines as opposed to positive displacement pumps.

The Roots' blower – Fig 1 – employs two identical 8 shaped rotors rotating in opposite directions inside a cylindrical casing. These rotors intermesh but maintain positive internal clearance via timing gears. Internal lubrication is unnecessary, so the air delivered is oil-free. Compression takes place by backflow every time a rotor tip uncovers the discharge point. Reasonable volumes of air can be displaced since the blower can be driven at relatively high speeds (*eg* synchronous speed of directly coupled electric rotors), but pressure generated is usually less than 1 bar from a single stage machine.

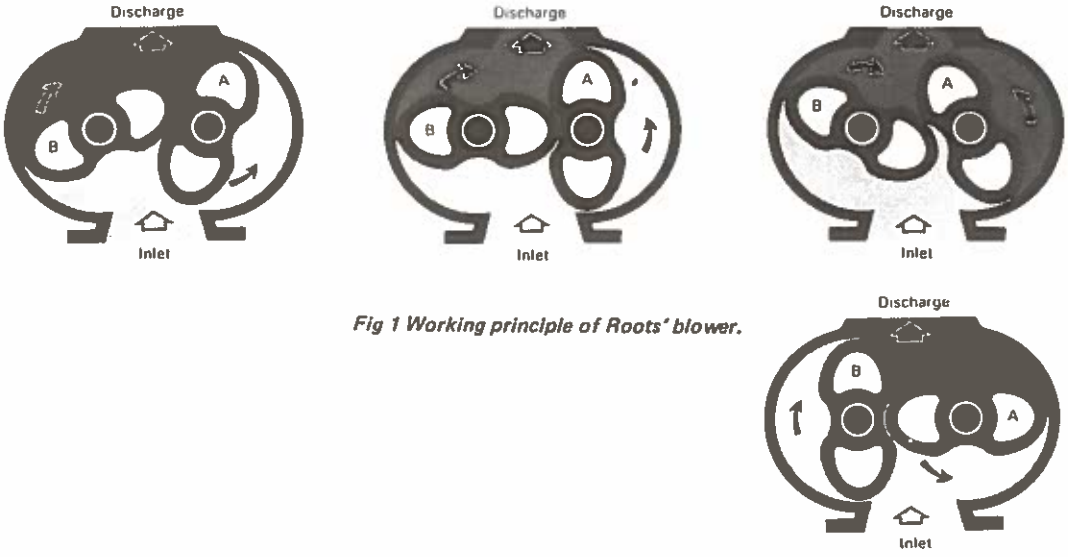
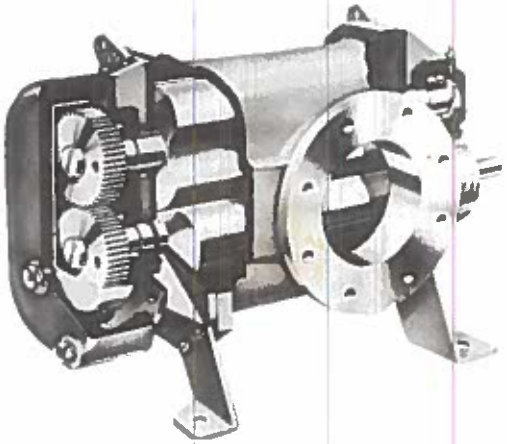


Fig 1 Working principle of Roots' blower.



Roots' type air blower. (Peabody Holmes Ltd)

The displacement (Q) can be expressed in the form of a general formula:-

$$Q = K(D^2 - C^2) \times W \times \text{rev/min}$$

where

D = rotor diameter
C = centre distance of rotors
W = width of rotors
K = a constant representing rotor efficiency, depending on the rotor geometry

A typical value for K is of the order of 1.5, although this will vary with rev/min (*ie* slip will tend to increase, and thus the rate of K will decrease with increasing rev/min).

Fan Blowers

A fan-type impeller consists of a number of blades mounted at a pitch angle and assembled on (or integral with) a hub, mounted on a driven shaft. Rotation generates a slip-stream in a similar manner to an airscrew, although the efficiency is very low compared with a true airscrew, since no attempt is made to provide a constant geometric pitch along the length of the blades (the blades may even have a constant pitch angle). Further, a relatively large number of blades may be used, with the resulting interference again reducing efficiency. At best, therefore, a simple propeller type fan can only generate a very low pressure, although it may be capable of moving a reasonable volume of air.

For a given blade size and shape, volume of air displaced will vary as rev/min, pressure as $(\text{rev/min})^2$; and horsepower input as $(\text{rev/min})^3$. For a constant speed, displacement varies as $(\text{diameter})^3$; pressure varies as $(\text{diameter})^2$; and horsepower input as $(\text{diameter})^5$.

Performance can thus be expressed quantitatively by the following formulas:

Volume = $k_v \times \text{rev/min} \times (\text{diameter})^3$
Pressure = $k_p \times (\text{rev/min})^2 \times (\text{diameter})^2$
Horsepower = $K_{hp} \times (\text{rev/min})^3 \times (\text{diameter})^5$

where

k_v = volume coefficient
k_p = pressure coefficient
K_{hp} = horsepower coefficient

Typical values of the fan coefficients are:-

$k_p = 0.65 - 0.90$
$k_v = 0.08 - 0.18 \times 10^{-6}$
$K_{hp} = 0.01 - 0.03 \times 10^{-9}$

The same fan laws and basic formulas also apply in the case of axial flow and centrifugal fans, only in these cases the values of the coefficients vary widely, depending on the specific design of the impeller and other empirical factors.

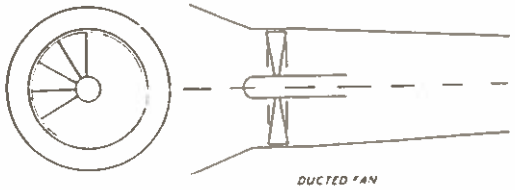


Fig 2

Ducted Fans

The performance of a simple propeller-type fan can be considerably improved by enclosing it within a close fitting open-ended cylindrical casing, producing a ducted fan – Fig 2. The design of the ducting is critical. It may be parallel sided or convergent-divergent (with a fan at the throat). Diffuser vanes may also be added. Much depends on the individual installation as to how efficient the fan is, although quite high pressure can be realized with suitably proportioned ducted fans.

For example with peripheral velocities of the order of 120 m/s (400 ft/sec) a pressure rise of 0.07 bar (1 lb/in^2) may be achieved although in practice this is seldom realizable because of the high noise level generated. Maximum peripheral velocities of the order of 75 m/s (250 ft/sec) are more usual, when, if higher pressure is required a multi-stage unit can be employed.

Particular advantages of propeller type fans are relatively simple construction, low cost and the ability to handle contaminated and even gritty air. Their short axial length, even with motor fitted, makes them suitable for installation directly in walls, or for compact installation elsewhere (with or without ducting). Their performance capabilities are limited to those of a true fan rather than a blower, although higher pressure levels can be realized with multi-stage units. They are widely used for straightforward ventilation, forced draught and cooling purposes and are normally quiet operating, running at low to moderate speeds. They may, however, pose both noise and vibration problems if run at higher speeds and higher efficiencies, or where the fan blades pass close to a stationary member. The latter problems can sometimes be overcome by adopting an asymmetrical fan form.

Axial flow fans are essentially a development of the simple propeller-type fan, developing the advantages of the latter but eliminating the disadvantages as far as possible. They are invariably ducted with the smallest possible clearance between the fan blades and casing consistent with safe operation. The smaller the clearance, the more effective is the casing in preventing recirculation of

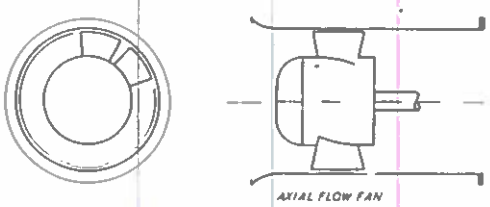


Fig 3

the air at the blade tips and loss of efficiency. The sides of the casing are normally parallel, although entry may be convergent, and relatively long to completely enclose both the fan and motor. Particular attention is given to the aerodynamic form of the impeller blades and to streamlining the through-flow. As a consequence an axial fan can generate a higher pressure than a propeller-type fan and has a higher efficiency. It is still, however, essentially a low pressure machine, with high volumetric capability (Fig 3).

Critical parameters are the aerofoil form of the blades, number of blades, blade angle and area, overall diameter and hub ratio – in addition to tip clearance and through-flow design as already mentioned. For low pressure working, the hub ratio is relatively small (*ie* a small diameter hub with long blades). At the other end of the scale a higher pressure axial fan (blower) would have a large hub ratio and short blades, possibly of quite different aerofoil section. More attention may also be paid to the design of the fairings to streamline the flow. Guide or diffuser vanes may also be fitted downstream to straighten out the flow and improve pressure development.

Noise can be a problem and can limit the practical maximum speed of the fan. In this case the critical parameter is the entry velocity of the air, a nominal limit being about 60 m/s (200 ft/s) above which velocity the noise can be intolerable. The presence of guide vanes can also cause noise problems by setting up local oscillatory flow, which can generate intense high pitched noise.

To increase the pressure available from an axial flow fan a two-stage unit may be employed. The stages may be driven by a common shaft (*eg* by the same motor with a shaft at each end), or be contra-rotating. In the former case, fixed flow straightening vanes are necessary between the two impellers, so that the second impeller can operate efficiently. No inter-vaning is necessary with contra-rotation, although the mechanics of the drive are far more complicated. In practice, the simplest solution is often to drive the two fans by separate motors. The impeller form may also be modified slightly to provide forced vortex flow.

A further development is the bifurcated coaxial fan where the motor is isolated from the downstream flow by taking this flow through two separate ducts – Fig 4. This can be particularly useful where the air being handled is contaminated, corrosive or excessively hot, and could cause damage to a motor exposed to it. It also places the motor in a very accessible position for inspection and maintenance. Alternative methods of isolating the drive motor include the use of V-belt drives, suitably faired, and mounting the fan at a bend in the ducting system, where a straight drive shaft can be taken through the wall of the bend to an externally mounted motor.

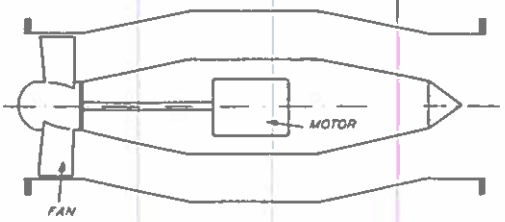


Fig 4

Radial Flow Fans

Radial flow impellers have similar characteristics to centrifugal compressors. The inducted air is turned through a right angle and ejected radially from the impeller. This provides greater pressure generation. The volume of air moved depends largely on the form of the impeller. Three basic blade forms are employed on blowers:-

(i) forward-curved blades (squirrel cage impeller)

(ii) backward-curved blades

(iii) straight radial blades

The forward-curved bladed impeller — Fig 5 — has a pressure-generating potential some five times greater than that of an axial flow fan and because of its form, with a large number of scoop-shaped blades usually of generous length, also has a high volume capacity. It is thus particularly suitable for handling a large volume of air where higher pressures are required than could be given by an axial flow fan. However, for the same capacity it will be much larger in diameter and more costly. Also its efficiency tends to be relatively low and restrictions may be placed on operating speed because of the noise level generated. This in turn can reduce the effective or usable capacity of the blower. They are used only when the discharge duct is short, since the high velocity at the discharge of the volute should not be converted to pressure.

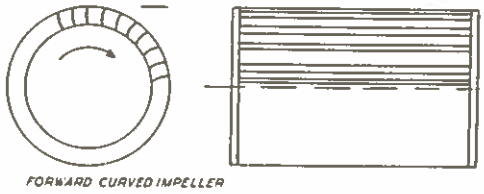


Fig 5

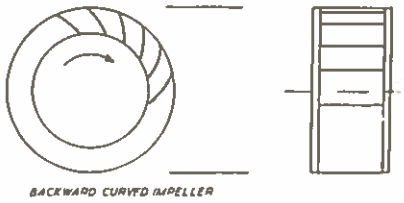


Fig 6

The backward-curved impeller (Fig 6) is also a high volume machine, capable of generating high flows at lower speeds than the forward-curved fan. Size for size, it could handle the same volume of air at an appreciably lower speed, and thus operate with a lower noise level. The latter advantage is further enhanced by the fact that the flow through the blades remains relatively streamlined, although much depends on the aerofoil form of the blades. True aerofoil blades produce the smoothest flow and highest efficiencies, but result in the highest cost impellers. Flat plate blades may be used on simpler, less expensive designs, still capable of giving a good performance with high efficiency. The basic configuration is still sufficiently versatile to enable it to be suitable for working over a wide range of sizes and specific speeds and so is a widely favoured type for blowers. Pressure capabilities are also high.

The pressure generated by either type of centrifugal impeller is generally sufficient for all blower duties, thus more than one stage will not usually be required. Multi-stage units are, however, used as compressors. Impellers with forward-curved blades are generally unsuitable for parallel operation.

The radial bladed impeller — Fig 7 — is essentially a high pressure low capacity form (virtually the opposite characteristics to an axial flow fan). It has a relatively low efficiency. Its chief advantages are the simple construction and the open form of the impeller, which make it particularly suitable for working with dusty or gritty air — hence its common application as an exhaust fan in

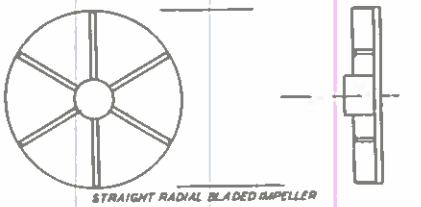


Fig 7

heavily contaminated atmospheres. If necessary the impeller can be made even more non-clogging by fitting a backplate to prevent fibrous material collecting at the back of the hub, or be protected against a corrosive atmosphere by simple coating treatment.

Some performance comparisons between axial flow fans and the three basic forms of centrifugal blowers are summarized in Figs 8 and 9. Fig 8 shows the characteristic horsepower and efficiency curves, and Fig 9 static pressure curves. It should be noted that these are related to 100% performance of the individual machines and that no quantitative comparison between the different types can be drawn for pressure or volume.

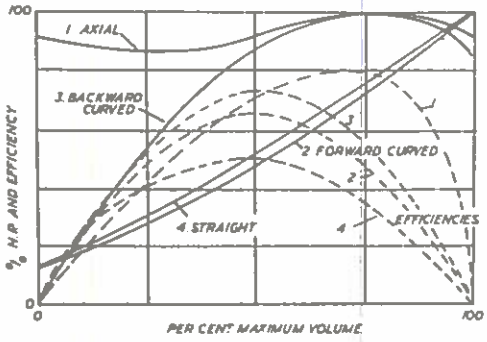


Fig 8

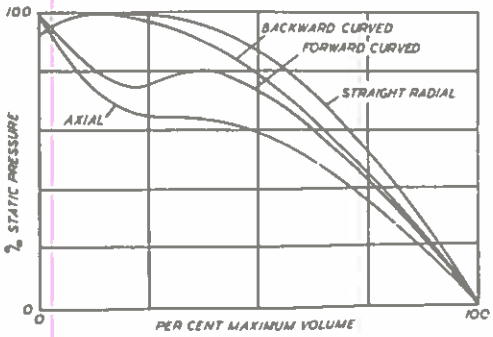


Fig 9

Choice of Type

For any application it is obviously desirable that the pressure and volume requirements be met with the highest possible machine efficiency. A convenient parameter for relating machine performance parameters and efficiency is *specific speed*, calculated as:-

$$\text{Specific speed } N_S = \frac{\text{rev/min} \times (\text{volume})^{0.5}}{(\text{static pressure})^{0.75}}$$

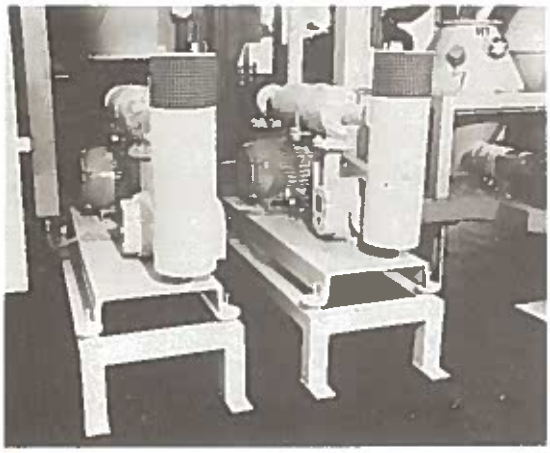
Each type and size of machine will have a specific speed range over which its efficiency will be around a maximum, and initial selection of a machine type can be made accordingly – see Table I.

The performance characteristics of the selected type must then be analyzed to see if it can provide the actual pressure and volume required, at a practical number of revolutions per minute.

Propeller-type fans are most efficient operating at high specific speeds, consistent with high rotational speeds, and very low pressures. Axial flow fans can operate efficiently at lower specific

TABLE I – OPTIMUM SPECIFIC SPEEDS FOR FANS, BLOWERS AND POSITIVE DISPLACEMENT MACHINES

Machine	OPTIMUM SPECIFIC SPEED RANGE	
	Metric Units	English Units
Propeller-type fans	600 000 – 2 500 000	100 000 – 400 000
Axial flow fans	300 000 – 2 000 000	50 000 – 300 000
Centrifugal impellers:		
Forward-curved, single suction	55 000 – 25 000	9 000 – 40 000
Forward-curved, double suction	185 000 – 425 000	30 000 – 70 000
Backward-curved	60 000 – 500 000	60 000 – 500 000
Straight radial blade	25 000 – 120 000	4 000 – 20 000
Positive displacement (compressor)	1 000 – 8 000	6 000 – 50 000



Two size OM type HR80 Holmes positive displacement air blowers installed at the Batley Works of Fox's Biscuits Ltd. The blower supplies the necessary oil free air for the pneumatic conveying of sugar. The pneumatic conveying system was designed and installed by Atlas Equipment (London) Ltd.

speeds, but maintain good efficiency into the region also covered by propeller-type fans. They are essentially high volume/low pressure devices, with again the emphasis on high rotational speeds.

The specific speed range of centrifugal blowers is less clearly defined, since this is influenced by individual design geometry, particularly in the case of forward-curved bladed impellers. Here two separate ranges are shown, one for impellers with single suction and the other for impellers with double suction. The two together cover a similar range to that of the backward-curved bladed impeller. The straight radial bladed impeller achieves its best efficiency at much lower specific speeds, although it overlaps the lower part of the other centrifugal types.

A generalization which can be drawn from this is that a high specific speed is consistent with high volume, low pressure delivery; a low specific speed is consistent with high pressure, low volume delivery. Positive displacement machines also fall into the latter category, with specific speeds extending down even lower than the straight radial bladed centrifugal blower.

Airmovers

THE JETFLOW Airmover is a device which converts a small quantity of low pressure air into a large volume of fast moving air, its principal application being auxiliary ventilation of confined areas, or localized ventilation. Since the device works without power (other than a supply of low pressure air) or moving parts, it can provide safe ventilation in potentially explosive atmospheres.

The principle of operation is shown in Fig 1. A supply of primary air enters the manifold where it expands to induce a depression at the throat of the venturi section and also attaches itself to the current aerofoil surface by Coanda effect. Secondary air from the surroundings is induced into the throat of the manifold, where it mixes with the primary air and accelerates with it through the divergent tube. The result is a high velocity mixture of primary and secondary flows emitted from the end of the tube, entraining more surrounding air.

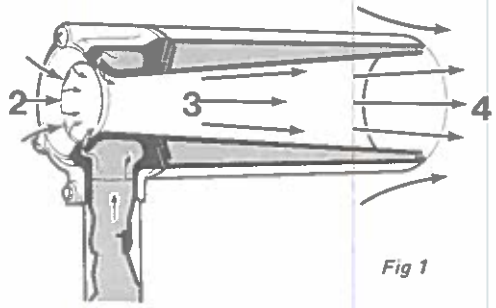
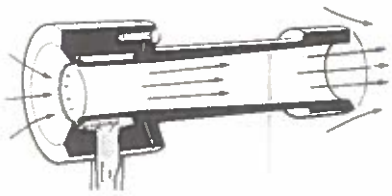
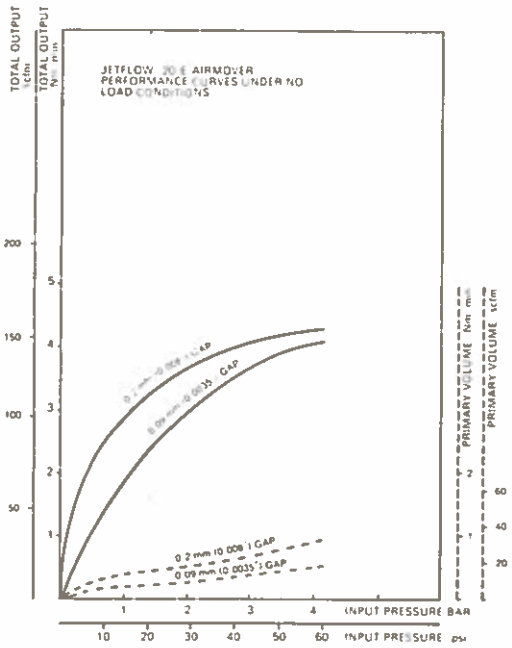


Fig 1

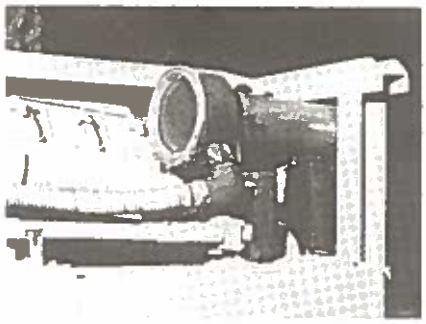
1 – Primary supply enters the manifold via annular gap and accelerates over the aerofoil.
2 – Secondary supply is induced into the throat of the manifold.
3 – Mixing, cooling and diluting in the divergent tube.
4 – Expelled at high velocity entraining surrounding air.



Olin 'Jetflow' 20 air mover.



Performance characteristics of Jetflow 20-E Airmover.



Typical installation of a 'Jetflow Airmover' on a diesel powered mobile vehicle. Engine exhaust is connected directly to the inlet manifold providing primary supply to cool and dilute the toxic exhaust gases.

Jetflow Airmovers can be operated singly, in series or in parallel and they perform extremely well when connected to ducting. There is, however, a limit to the resistance they will overcome but at best the Airmover will generate a static pressure (suction) of about 112 m/bar (45" W.G.). The resistance to airflow is determined by the length and diameter of the ducting and the frictional characteristic (K factor) of the duct lining.

Adequate Airmover performance depends on the correct application of these variables:-

(a) Gap Setting.

(b) Pressure of primary supply in manifold.

(c) Volume of primary supply available.

The Jetflow Airmover will operate effectively at manifold pressures varying from 0.04 bar (0.5 lb/in^2) to 3.9 bar (55 lb/in^2). It is recommended that the maximum manifold pressure does not exceed 4.2 bar (60 lb/in^2). In many applications most efficient operation is attained within the range 0.35 bar (5 lb/in^2) to 1.4 bar (20 lb/in^2).

Vacuum Pumps

THE TWO major parameters in the design of a vacuum system are the rate of evacuation required and the ultimate pressure to be realized and held. These parameters determine the size and number of pumps required. This may involve the use of *backing pumps* capable of pumping from atmospheric pressure downwards, followed by further pump(s) which only begin to operate below a certain limiting pressure level.

The majority of backing pumps are of the mechanical type, although fluid jet pumps, condensers, liquid air traps and cryogenic pumps will work from atmospheric pressure down. Mechanical pumps are widely used for producing coarse vacuums, and can extend into the medium vacuum range by adopting a second stage. The first stage is then effectively acting as a forepump for the second stage in pumping off the quantities of gas removed by the second stage against the fore-pressure provided by the first stage.

TABLE I – TYPICAL PERFORMANCES OF MECHANICAL VACUUM PUMPS

Type	Design	Minimum Partial Pressure (approx)
Reciprocating	Dry – single-stage	10 torr
	Dry – double-stage	1 torr
	Wet	20 torr
Rotary vane	Single blade – single-stage	5×10^{-1} torr
	Single blade – double-stage	2×10^{-5} torr
	Double-blade – single-stage	5×10^{-5} torr
	Double-blade – double-stage	2×10^{-5} torr
	Multi-blade	5×10^{-1} torr
Rotary plunger	Single-stage	5×10^{-3} torr
	Double-stage	2×10^{-5} torr
Twin rotor	Roots – single-stage	100 torr
	Roots – double-stage	10 torr
	Roots – type booster	10 torr down to 10^{-3} torr
	Lysholm	2×10^{-1} torr
	Northey	2 torr

TABLE II – VACUUM PUMPS – TYPES AND CHARACTERISTICS

Type	Normal or Typical Operating Range	Remarks
Reciprocating, dry type	Single-stage – down to 10 torr Two-stage – down to 1 torr	Vertical or horizontal Single-, twin- or multi-cylinder
Reciprocating, wet type	Single-stage – down to 5 torr	Slow running speeds normal
Vane, dry type	Down to 5×10^{-1} torr	With or without gas ballast
Vane, oil sealed	Single-stage – down to 5×10^{-2} torr Two-stage – down to 5×10^{-5} torr	
Meshing rotor	Single-stage, down to 1 torr, depending on type Two-stage, down to 10^{-1} torr	With or without gas ballast
Rotary plunger	Down to 10^{-2} torr	With or without gas ballast
Molecular drag, finned rotor	10^{-1} to 10^{-5} torr	Require fore-pump
Molecular drag, plain rotor	10^{-1} to 10^{-6} torr	
Steam ejector	Single-stage – down to 26 in Hg Two-stage – down to 29½ in Hg Three-stage – down to 29¾ in Hg Four-stage – down to 1 torr Five-stage – down to 10^{-1} torr	Normally employed with intermediate condensers
Liquid ring	Down to 50 torr	Single- or two-stage Liquid used is generally water. Remove condensable vapours.
Water ejector	Down to 50 torr	Larger than steam ejector for similar duty.
Vapour booster	From 10 to 10^{-2} torr	Combines diffusion pump backed by ejector pump
Diffusion, oil	Non-fractionating – from 10^{-1} to 10^{-6} torr Semi-fractionating – from 10^{-3} to 10^{-7} torr Fractionating – from 10^{-2} to 5×10^{-8} torr	Require fore-pump for backing pressure
Diffusion, mercury	From 10^{-2} to 10^{-8} torr	Regular freezing traps may be preferred for oil-free systems
Cryogenic	Down to 10^{-10} torr	Condensation pumps employing liquefied gas refrigerant
Chemical (absorption)	Down to 2×10^{-2} torr	Alumino-silicate chemical compound at –200°C (liquid nitrogen coolant)
Sorption	Down to 10^{-3} torr	Molecular sieves at *circa* –200°C (liquid nitrogen coolant).
Getter-ion (evaporative ion)	5×10^{-4} to 10^{-11} torr	Usually titanium or zirconium getter. Outgassing reduced by uniform baking prior to pumping (*eg* at 450°C).
Cold cathode (penning pump)	5×10^{-2} to 10^{-11} torr	Outgassing reduced by uniform baking prior to pumping (*eg* at 500°C)

Mechanical Pumps

In the case of mechanical pumps, the following is an empirical formula wich can be used when initial evacuation time is not critical:-

$$S_m = 4\,V^{2/3}$$

where

S_m = the minimum mean speed of the pump between atmospheric pressure and operating vacuum in l per minute

V = the volume of the system in l^3

Where evacuation time and the vacuum level to be attained are critical, the formula:-

$$S_e = \frac{2.3V}{t} \log_{10} \frac{p_1}{p_2}$$

should be applied, where

S_e = mean effective speed in pump in l per minute

V = volume in l

t = permissible evacuation time in minutes

p_1 = initial system pressure in torr

p_2 = final system pressure in torr

Pump capacity necessary to maintain the required vacuum level against air leakage only can be estimated from the following empirical formula:

$$S_e = \frac{Vp_i}{6p}$$

where

S_e = effective speed of pump at operating pressure in l/minute

V = volume of system in l

p = operating pressure in torr

p_i = pressure rise due to air in leakage torr/hour (measured by pressure rise test and/or specified as leak-tightness standard).

In a practical system account must also be taken of gases evolving from the process or system.

'Blank-Off' Pressure

It is normally recommended that in selecting vacuum pumps, the minimum 'blank-off' pressure capability of a pump should be of the order of a half to one decade lower than the ultimate operating pressure to be attained by the pump. Thus a rotary piston pump with a 'blank-off' capability of 0.01 torr would be suitable for use on a system requiring an ultimate operating pressure above 0.05 to 0.2 torr. Increasing the margin between 'blank-off' and operating pressure is advantageous but decreasing it below the recommended figure will result in reduction of system reliability.

Many single stage rotary piston pumps are capable of reaching 'blank-off' pressures of 0.010 torr or lower, which represents a compression ratio capability of the order of 100000 to 1. In terms of operating pressure range, such pumps perform efficiently between 75 torr to 0.075 torr and thus have operational compression ratios ranging from 10 to 1 to 10000 to 1.

The efficiency of the pump can be estimated from the following formula:

$$F_C = (F_1 \times F_2)^{0.5}$$

where

F_C = combined pump efficiency factor

$$F_1 = \frac{Sav}{41.63\ kW}$$

$$F_2 = \frac{2.88 \log_{10} p}{4.88}$$

where

Sav = mean speed of pump from atmosphere to pressure p.

kW = installed power in kilowatts of driving motor(s).

p = lowest pressure obtained in torr

Gas Ballast (Fig 1)

If water vapour (or other vapour) is likely to be present in the system it will cause trouble by condensing during the compression cycle of mechanical pumps, contaminating the oil circulating in them. On entering the high vacuum side, this condensation will re-evaporate, reducing the ultimate pressure and pumping speed.

This can be overcome by using the technique of gas ballasting on the backing pump in the system.

Gas ballasting involves the ingestion of a small quantity of gas (usually air) at a constant rate of mixture, preventing condensation during compression provided the sealing oil temperature is above the saturation temperature of water vapour at the reduced partial pressure. Normally, for effective gas ballast, the oil temperature should be maintained at 60°C to 70°C (140°F to 160°F) where its lubricating and sealing properties are not impaired due to reduction in viscosity.

Vane pump working with gas ballast (water vapour ejected with gas ballast).

Vane pump working without gas ballast (condensed water vapour remaining over a certain level if partial evacuation is needed).

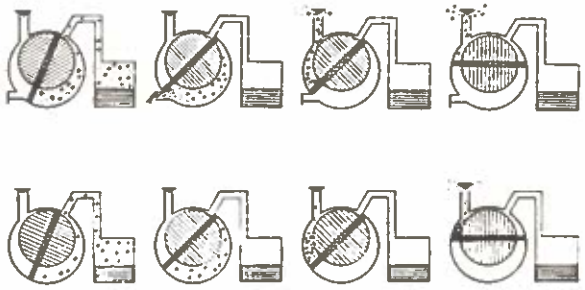


Fig 1

The use of gas ballast limits 'blank-off' and operating pressure capability of the pump. Depending on the particular design and the characteristics of the pump, there is an upper limit to the pressure at which gas ballast is effective. This is normally in the region of 25–30 torr which limits the quantity of water vapour which can be handled by a particular capacity pump. Where large quantities of water vapour have to be handled, the mechanical pumps should be protected by a water-cooled or refrigerated condenser or trap, as condensing is the most efficient means of pumping water vapour.

Condensers

Condensers usually consist of a vacuum envelope containing tubes or parallel plates, which may be either water cooled or refrigerated. The most efficient vapour trapping path is achieved by correct positioning of the vapour inlet and outlet.

Condenser performance is independent of the capacity of the backing pump, which merely maintains the required low partial pressure. However, high permanent gas pressures must be avoided as they will blanket the cooled surface and reduce the condenser efficiency.

If very high water vapour pumping speeds are required, it is necessary to use a water cooled surface condenser. A condenser consists of a suitable designed tube unit which has a high cooling water velocity, and pumping speeds as high as 2–2.5 kg water vapour per m^2 condenser area per hour per degC mean temperature difference can be achieved. For lower partial vapour pressures, refrigerated condensers can be used. However, there is always the possibility that the vapour fluids may freeze.

Therefore, space must be left between the condenser tubes to prevent ice bridging across them and causing too great a temperature difference. The coefficient of heat transfer depends mainly on the thickness of the frozen layer; for water vapour which has formed an ice layer 6 mm thick, the figure is about 320 kcal/m^2 /degC. However, the temperature difference between the evaporating refrigerant and the ice also affects the coefficient, as this affects the turbulence and hence the rate of heat transfer.

A disadvantage of refrigerated condensers is that they must be defrosted periodically. Therefore, if continuous operation is required, two units must be installed in parallel; one condensing, the other defrosting.

An ideal pump/condenser combination which has excellent vapour handling characteristics, and will operate efficiently in a variety of conditions, is a two-stage mechanical system consisting of a rotary pump and a mechanical booster pump with an interstage condenser. Very fast vapour pumping is possible due to the fact that the interstage pressure is high, and vapour can be condensed and collected in a suitable receptacle.

Contamination of the primary low pressure pump is avoided by maintaining an interstage vapour pressure below that at which condensation occurs in the pump. The backing pump, which has a much smaller capacity, is protected by gas ballasting. Using this system it is possible to cope with water vapour with a partial pressure of up to 100 torr. Furthermore, solvents particularly those with a high vapour pressure, can be easily pumped, trapped and recovered.

Diffusion Pumps

Diffusion pumps as a type require a forepump for backing pressure, when they are capable of working from 10^{-8} down to 19^{-8} torr, depending on type. They are basically condensation pumps, using mercury or oil as the working fluid. Oil-diffusion pumps are generally preferred for industrial applications, and may be of either fractionating or non-fractionating type.

Diffusion pumps are manufactured both as individual pumps and pump sets incorporating a suitable forepump.

Oil diffusion pumps are capable of much greater pumping speeds than mercury pumps of similar size, because of the difference in molecular weight and molecular volume of the operating fluid. Oils also have a low vapour pressure compared with mercury and thus the pumping speed is relatively unrestricted. Also the use of a freezing trap can be avoided (usually necessary with a mercury diffusion pump to prevent mercury vapour from diffusing back into the vessel being exhausted), except in specific applications where it is absolutely necessary to ensure complete absence of oil vapour in the work-piece (*eg* as in the evacuation of envelopes of thermionic valves and similar devices).

The typical modern oil diffusion pump is capable of attaining pressures of the order of 10^{-6} torr without the aid of cold traps or baffle systems and maintains a high pumping speed over the majority of the working range. Apiezon oils are commonly employed as the working fluid, having a low vapour pressure and being extremely reliable in use. Construction is usually of metal with either water- or air-cooling. The jet system may be fractionating, semi-fractionating or non-fractionating, according to which is best suited to the particular application.

A certain amount of back-streaming is inevitable with a diffusion pump, although this may be largely reduced by fitting a baffle plate directly above the pump inlet. Some baffle plates incorporate a valve as well as a baffle so that the pump can be completely isolated from the vacuum system if necessary. The use of a baffle plate, however, automatically means a reduction in pumping speed and so where one is employed it is most important that the design offers the best possible compromise between reduction of back-streaming entirely and where it is necessary to reduce back-streaming to an absolute minimum refrigerated chevron baffles or cold traps have to be employed.

For applications where the slightest trace of oil vapour in the vacuum system must be avoided, mercury diffusion pumps may be used. To eliminate mercury vapour from the vacuum system (although this may be acceptable in some cases, where oil vapour would not), very low temperature traps are employed, using liquefied gases as coolants.

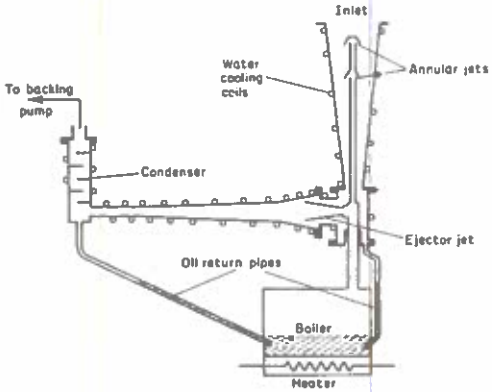
Vapour Booster Pumps

Vapour booster pumps fall between the conventional diffusion pump and a rotary pump as far as general performance is concerned. They comprise, usually, a diffuser pump (one or more stages) backed by an ejector pump or ejector stages.

Fluid Jet Pumps

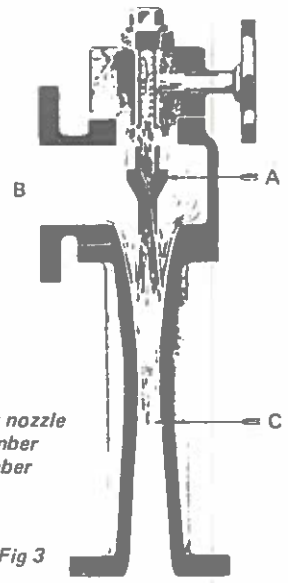
Fluid jet pumps work on the principle of introducing a high-velocity fluid jet (liquid or vapour) into a mixing section where it meets the gas to be evacuated and entrains that gas by turbulent mixing. Some of the momentum of the jet is thus imparted to the gas which is carried down into a diffuser section and thence to a condenser if necessary, for separation. Where the operating fluid is a liquid, separation can be achieved in the diffuser, and where the operating fluid is a gas usually in a condenser, although the complete system may comprise two or more stages, with or without intercooling, depending upon the performance required. The usual types may be grouped as water jet pumps (fluid), steam ejector pumps (vapour) and vapour-booster pumps. The two former are essentially similar in principle and application whilst the vapour-booster pump comprises a two-stage system with a condenser — Fig 2.

The steam ejector pump has the most wide-spread industrial application and has, in fact, largely replaced other types of vacuum equipment in certain fields (*eg* for use with power plant condensers) because of its simplicity, reliability of operation and low maintenance costs. It can also be



Vapour booster pump

Fig 2



A – divergent jet nozzle
B – suction chamber
C – mixing chamber

Fig 3

used to handle corrosive or extremely hot gases, *etc*, since many materials can be adapted for its construction – *eg* the major pump elements can be made from carbon or refractory materials, glazed ceramics, *etc*, to resist corrosive attack. It may be termed variously a jet pump, ejector, augmenter (or vacuum augmenter), or thermo-compressor, depending on the duty performed. When used to produce and maintain a condition of vacuum by the removal of incondensable gases it is correctly termed an *ejector.* If used to compress vapours from a condition of high vacuum to a lower vacuum at which condensation can take place, it is correctly termed a vacuum augmenter.

The design of all steam jet pumps is essentially similar and follows the form shown in Fig 3. Steam is introduced via a divergent nozzle into a converging section of the body. The body section also incorporates a suction branch connected to the vessel or system to be evacuated and gases drawn in through this opening are entrained with the steam jet and carried down with it into the diffuser section or compression tube. During this process of entrainment the steam jet loses a proportion of its energy which is imparted to the aspired gases. In the diffuser section the kinetic energy of the mixture of steam and gases is re-converted into pressure energy. The degree of compression which can be achieved is proportional to the amount of kinetic energy each unit mass of mixture has when it enters the diffuser and thus, basically, to the quantity of steam passing through the jet nozzle. The normal range of single-stage ejectors embraces a steam consumption ranging from about 50–60 up to 8000–9000 lb per hour.

The compression ratio of the ejector is defined as the ratio of the absolute pressure of the steam-gas mixture at the discharge end to that of the absolute pressure of the aspired gas at the suction branch. Compression ratios of up to about 15:1 are practical with simple ejectors, although it is more usual to work to a lower figure in order to achieve better economy in steam consumption. Thus the single-stage ejector produces economic operation down to vacua of about 26 in. For higher vacua a two-stage or multi-stage unit is normally preferred.

When two or more stages are employed it is usual to employ a condenser between each stage so that the steam of the first stage is condensed out and withdrawn before reaching the next stage.

Thus each succeeding stage has only to deal with the incondensable vapours remaining. The usual method of cooling is to employ a cold water spray when the steam is condensed by direct contact with the cooling water, whilst at the same time the incondensable gases are cooled and reduced in volume before being drawn into the suction branch of the following stage ejector.

Three-stage ejectors are usually provided with an intermediate condenser for the first stage (although this may be omitted on smaller units) and a further condenser following the second stage. Four-stage units seldom require a condenser following the first stage since the amount of steam consumed from this stage is small and can be efficiently compressed by the following stages. Five- and six-stage units follow on the lines of a four-stage unit, with the addition of one or two non-condensing stages. The bulk of industrial applications are covered by one-, two- or three-stage injectors.

Water-operated air ejector pumps operate on a similar principle to the steam ejector except that the velocity of the water jet is due to pressure alone rather than expansion and is therefore usually much lower (steam jet velocities of over 1200 m/sec (4000 ft/sec) are commonplace in steam ejectors designed for high vacua). For a similar duty, the water ejector pump is considerably larger in order to deal with a similar volume of gas (normally air). Initially the air is in contact with the outer surface of the water jet only, until subsequently entrained in the cone.

See also chapter on *Vacuum Techniques*.

SECTION 5

Techniques

Noise Control

ALL PNEUMATIC machines, tools and devices tend to be noisy because relatively high pressure air is exhausted direct to the atmosphere, producing a choked jet stream. Noise level may further be raised by the mechanical noise generated by the device itself (particularly in the case of percussive tools). Basic solutions are to silence the exhaust and apply some form of sound deadening or sound absorption to the tool itself.

Appliance Noise

Pneumatic tools and appliances basically exhaust compressed air directly to atmosphere. Where the pressure ratio at the exhaust exceeds 2:1 the exhaust flow will be supersonic. Shock waves are generated and a considerable degree of turbulence with the sudden breakdown of the shock waves and mixture of high velocity air with surrounding static air. Noise reduction can only be effected by (i) eliminating shock wave formation and (ii) reducing the exit velocity as much as possible to minimize the degree of turbulence. Both depend on the pressure ratio involved.

The most effective way of reducing the noise is to allow the air to escape into an expansion chamber in order to reduce the pressure ratio to below 2:1 before it finally escapes into the air. Exhaust flow will then be subsonic and restricted to pulses of flow and turbulent mixing.

Silencer fitted to Airmac MK20 'Super Silent' Breaker.

Three further problems then exist. The flow into the expansion chamber will still be supersonic, with the formation of shock waves. Suitable attention must, therefore, be given to the expansion shocks so formed. Noise from the final outlet will be primarily dependent on the escape velocity, which is in turn dependent on the final pressure ratio and the area of outflow. Further reduction in noise is thus possible by increasing the outflow area, so that the flow is diffused. The final requirement is that the complete device, or silencer, does not impart excessive back pressure to the outflow, which could affect the operating efficiency of the device.

The simplest type of silencer consists of a simple expansion chamber attached to the exhaust port of the device — Fig 1. This provides a volume in which the pressure ratio can be reduced to a suitable figure, ie P2 less than 1.9 times atmospheric pressure, when the exit flow will be subsonic. The final noise energy will be proportional to AV_2^8, from which it is obvious that anything reducing the value of V_2 will have a substantial effect on the noise energy level. Some improvement can often be realized by the simple practical expedient of attaching a length of hose to the silencer outlet to act as a second expansion chamber, further reducing the value of V_2.

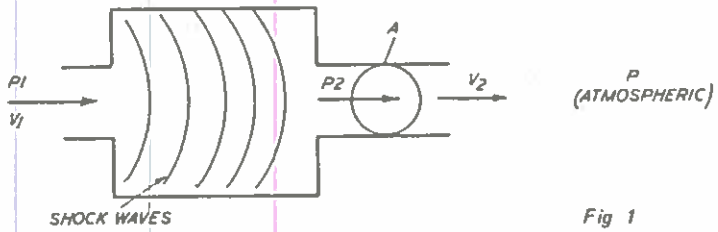


Fig 1

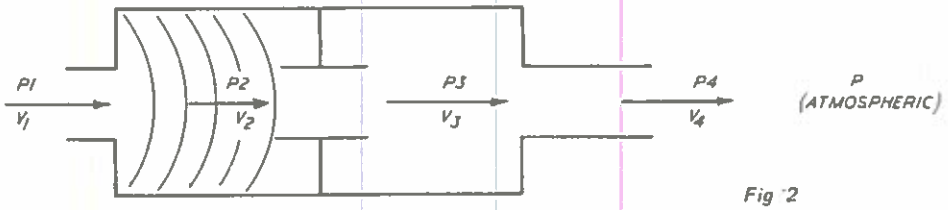


Fig 2

Silencers of this type may be in the form of simple attachments to fit on the exhaust port of a cylinder or valve, or in specific cases may be incorporated in the design of the tool or device. Noise reduction of up to 10 dB(A) is generally possible, with a suitable chamber volume. Further improvement can be realized by using a two-port chamber, and/or combining an expansion chamber with a diffuser. Here savings of up to 25 dB(A) are sometimes possible, although the figure achieved is usually materially less (Fig 2).

Two-chamber expansion provides better control of expansion shocks, particularly if the first chamber can be enclosed. This will prevent any expansion shocks formed from passing directly out of the final port. Also final expansion through the exhaust port can take place isothermally, or nearly so, because of the low pressure ratio involved. This is particularly useful, since it eliminates the possibility of ice build-up in the exhaust chamber or exhaust port — a problem which can arise with simple single-chamber exhaust silencers. This, in fact, is the main limitation of simple silencers. The sudden drop in temperature of the air on expansion deposits water vapour which can

freeze around the exhaust port or exhaust passages, or even block the silencer completely. In practice, however, complete blockage is not likely because of the scouring action of the high velocity air passing through the chamber, provided the flow path is simple. Icing up of the first chamber is also a possibility in the case of two-chamber expansion. Here, however, the design can allow for the fact that should the first chamber become iced up, flow to the second chamber will not be restricted. In that case the exhaust cannot become restricted by icing up, only the efficiency of the silencer reduced.

Percussive tools are inherently noisy in operation, since the majority of the expansive energy of the air is dissipated in the exhaust at each end of the stroke; a piston impacts against an anvil block at a frequency of the order of 20 times per second and a high intensity stress wave is transmitted down the drill rod at a similar frequency. The typical noise spectrum comprises a number of discrete frequency peaks spread over a wide range with a high overall level of the order of 100 dB — see Figs 3 and 4. Basically, however, about 80% of the noise comes from the exhaust, and most of the remainder from the rod. Simple jacketing, without any attempt to silence the exhaust, would thus be singularly ineffective.

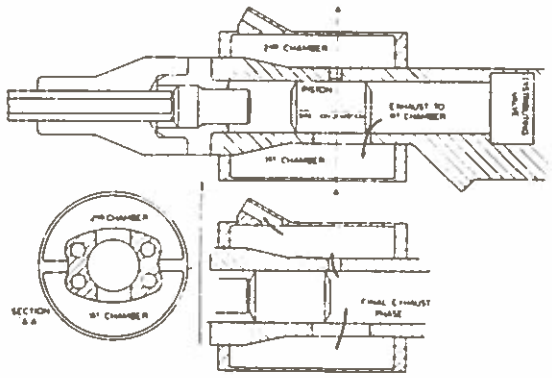


Fig 3

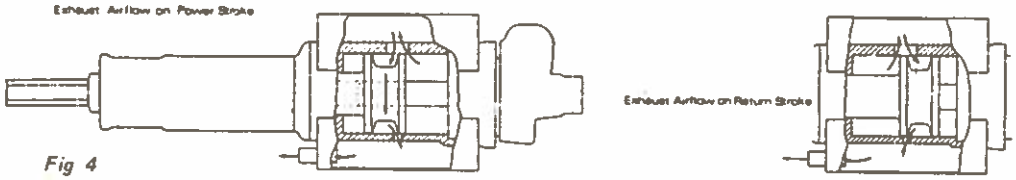


Fig 4

Fitted jackets or mufflers provide a means of silencing an existing (unsilenced) tool without any structural modification. Jacket-type mufflers (eg Pfister and similar) are also generally lighter and comprise, basically, a lightweight jacket of PVC coated terylene lined with profiled foam rubber, which is reinforced adjacent to the exhaust positions. They are held in place with cords and straps.

Rigid, detachable mufflers (basically exhaust silencers) may also be produced by manufacturers to fit individual standard tools, although in this case the tool must be designed initially with provision for accommodating a matching silencer. The simpler solution of producing an exhaust silencer to fit on to the exhaust in the form of an extension piece is less attractive because of its vulnerability. Also the efficiency of simple single-chamber exhausts can be relatively low, and variable, if icing-up is experienced.

Particularly good results have been obtained on road breakers with both exhaust and steel muffling, and the combination of a silenced portable compressor with a fully muffled breaker is likely to meet most of the objections from the public when they are used in a built-up or residential area. It is unlikely that any further reductions of appreciable magnitude can be obtained in this equipment in the near future. It has been suggested that hydraulic breakers which are coming onto the market now are much quieter than the corresponding pneumatic ones, but tests show this not to be so.

Simple exhaust systems applied to pneumatic tools are generally based on the addition of a collector ring around the body of the tool and encircling the original exhaust openings. This is suitably sealed to the body to form a complete expansion chamber. Some further silencing may be achieved by the addition of a length of hose to the exhaust outlet of the silencer, the collector ring then acting as a first expansion chamber with subsonic flow through the hose and to the final exit. The noise reduction achieved by these methods on a pneumatic hammer is shown in Fig 5.

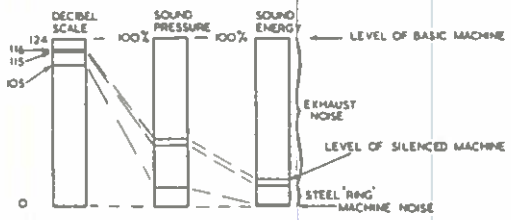


Fig 5 Comparison of the decibel scale with sound pressure and energy for an experimental silenced rock drill. (Holman Bros.)

Tool Steels

Sound reduction treatment of steels remains a particularly difficult problem. Piston and anvil noise is, to some extent, alleviated by the use of a padded jacket, when a noise reduction of up to 10 dB(A) may be achieved, although the figure is usually less. There is also user-objection to the greater bulk of a padded tool.

The direct approach is the use of a high damping material for the rod, although unless the material has directional properties this will inevitably lead to a loss of effectiveness of the tool point.

An alternative solution which has been developed by BroomWade is the fitting of an elastomeric damping device located at a critical point on the steel. The object of this is to reduce mechanical vibration and recoil, and hence the noise radiated. A comparison under similar conditions, between a standard breaker using a standard steel and a muffled breaker with elastomerically damped steel has shown a noise reduction in the latter case as high as 10 dB(A).

Fan Noise

The noise generated by a fan tends to be directly proportional to the static pressure developed and the bhp absorbed by the fan. Other major parameters covered are:—

(i) Air velocity — responsible for self-noise.
(ii) Turbulence — responsible for frequency content of noise.
(iii) Operating rev/min — responsible for mechanical noise (bearing and motor noise).

An empirical formula for estimating the fan sound power level direct from pressure, volume and input power is:—

$$SWL = 90 + \log_{10} hp + 10 \log_{10} P$$

or

$$SWL = 55 + 10 \log_{10} q + 20 \log_{10} P$$

or

$$SWL = 125 + 20 \log_{10} hp - 10 \log_{10} q$$

where SWL = overall sound power level of the total noise spectrum (31.5 Hz to 8 kHz) transmitted to the associated ductwork in either upstream or downstream direction (measured in dB re 10^{-12} watts)
hp = rated motor horsepower
P = static pressure (inches wg)
q = supply volume ft^3/min

Noise level and frequency content is also influenced by the number of blades in the fan. General characteristics of different fan types are summarized in Table I.

The noise spectra of fans vary with the type of fan, but in general they are typical of all aerodynamic noise in that the acoustic power generated over a wide spectrum rises to a shallow peak at a frequency determined by the type, size and duty of the fan. Centrifugal fans having either forward or backward curved blades tend to produce a spectrum which fails at approximately 5 dB per octave with increasing frequency, although there is a tendency for backward curved impellers to generate more energy at high frequencies and correspondingly less at low frequencies.

TABLE I — FAN NOISE PARAMETERS

Type of Fan	Air Velocity	Turbulence	Speed	Remarks
Propeller	Low	Low	Low	Inherently low noise level
Axial flow	Low* High†	Low* High†	High	*Low pressure types †High pressure types
Centrifugal				
Forward curved	High	High	High	Large number of blades
Backward curved	Low	Low	Low	
Straight radial blade	High	High	Medium	

Axial-flow fans have a much flatter spectrum and contain a much greater proportion of high frequency energy. Typical generalized spectra are given in Fig 6. The use of this chart can give octave band sound power levels of standard fan configuration to an accuracy of ±4 dB.

Not all noise generated by fans and transmitted along the ductwork comes from aerodynamic sources at the impeller. Some of the noise may be attributable to fan unbalance, bearing noise, structural resonance of the casing, or to noise generated in the motor, its couplings or belts. The fan noise may also be affected by its mode of installation and whilst the sound power generation

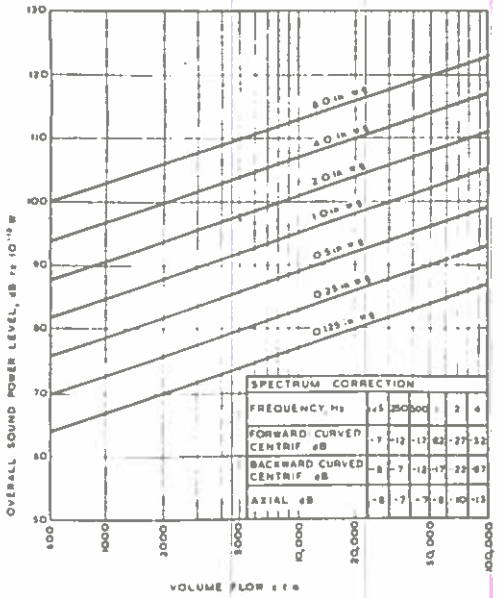


Fig 6 Chart for estimating octave band sound power generation by fans.

of a centrifugal fan does not seem to be greatly affected by variations in inlet and discharge arrangements, extreme care should be taken to ensure that axial flow fans are installed in accordance with the manufacturers' recommendations. Quoted sound power levels are generally applicable to the case where the axial flow fan is coupled to inlet and discharge ducts having the same diameter as the fan casing and a minimum straight duct-run upstream and downstream of the fan equal in length to two duct diameters. The rated sound power levels are also applicable to fans having coned inlets and inlet transformation sections, provided the included angle does not exceed 60^{o}

The octave band sound power level and the spectrum of fan noise vary not only with the speed and duty of the fan, but also with its type and individual manufacturer. A standardized procedure for the measurement of fan sound power output is laid down by British Standard 848, Part II, and most fan makers will be found to publish full details of noise generation by their products in accordance with this Standard.

Low Temperature Pneumatics

CONVENTIONAL PNEUMATIC systems are limited to operating temperatures above 0°C as compressed air is normally saturated with water vapour which would freeze at lower temperatures. This can be a problem with exposed outdoor pipeline systems in severe weather, for example. Where it is necessary to operate systems at sub-zero temperatures, two basic methods of approach are possible — (i) *drying* the compressed air before delivery; or (ii) *air dosage,* or injecting 'anti-freeze' into the compressed air. At the same time components themselves may need separate treatment (*eg* with local heaters or heated jackets) to ensure freedom from freezing and seizing up.

Dryers

The use of air dryers is well known and with modern types can be operated at economic levels on a continuous flow basis. Absorption-type dryers are capable of removing about 99.9% of the water content in compressed air, making it possible to operate a compressed air system down to about −70°C. The actual minimum working temperature achieved will depend on the performance and capacity of the dryer, and the limit to which it is worked. In fact, it is economic sense to work a dryer below its full potential water removal performance, adjusting the level for the amount of protection necessary. For example, something substantially less than 99.9% water removal will be adequate for working the system at, say −10°C.

Air Dosage

Air dosage is a much simpler, and usually cheaper method, injecting the compressed air with alcohol or a vaporizing spirit which effectively lowers the freezing temperature of the water vapour content. The effectiveness of air dosage, however, is limited by the maximum dosage of 'anti-freeze' that can be injected without diluting lubricant present to dangerously low levels. A typical maximum dosage is 10 drops per cubic metre of free air, which should be capable of lowering the minimum working temperature of the system to between −15° and −20°C.

Optimum point for injecting an anti-freeze dosage is immediately following the compressors, where the air is warm. This will promote rapid and complete vaporization of the injected fluid — Fig 1. In a small system, however, adequate performance can often be obtained by introducing the spirit dose in the downstream line or lines which are operating in a cold zone — Fig 2. When the

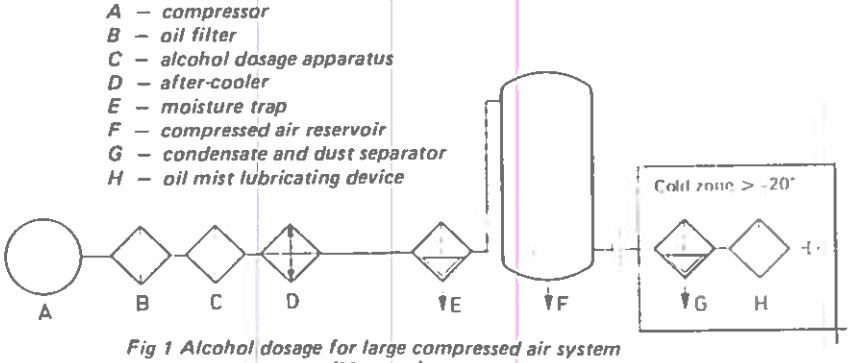


Fig 1 Alcohol dosage for large compressed air system (Mecman)

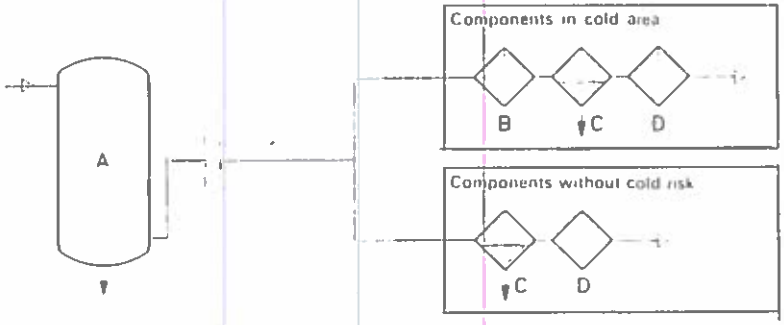


Fig 2 Alcohol mist dosage for small compressed air system (Mecman)

A – compressed air reservoir
B – alcohol dosage apparatus (oil mist lubricating device)
C – condensate and dust separator
D – oil mist lubricating device

supply feeds both cold and normal temperature service lines only those in the cold zone need dosage. Thus downstream dosage can be more economic where only part of the system has to operate in freezing temperatures. It also has the merit that the disadvantages associated with dosage are avoided in the 'normal temperature' service lines.

System Requirements

Air dosage or air drying provides protection to systems down to the temperatures indicated. This does not necessarily mean that the complete system will continue to function satisfactorily at these low temperatures. Some water vapour will still be present (particularly with spirit dosage), and ice may still form at points where local expansion of the compressed air produces a further drop in temperature. Also water vapour present in the surrounding air can turn into ice in contact with low temperature parts in the system – eg outlet ports. Localized heating of components may, therefore, be necessary, to maintain satisfactory minimum temperature working.

In less severe applications, localized heating may prove satisfactory on its own. Thus, where an air line has to pass through a cold zone, heating by a jacket or similar means may be sufficient to

maintain the air supply above dew point. If valves are located in cold zones, these can be protected from freezing by containing them within a thermally insulated enclosure with simple internal heating (eg provided by a low or moderate wattage light bulb). If localized heating is considered too complicated or unjustified, then alcohol dosage, applied at the critical point of the system where freezing troubles occur, may offer a simpler solution.

The main risk of ice formation in pipes occurs just after throttling points. If the system is intermittent in operation the quantity of ice formed may be insignificant and melt during the off times without giving any trouble. Icing problems may well show up on such a system operated at a faster rate or for longer periods of time because of restricted melting times and consequent ice build-up.

Operating at low ambient temperatures, but above freezing point, these critical areas subject to freezing may well show up by the formation of ice on the outside of the pipe. This in itself is not necessarily harmful in a continuously operated system; but with intermittent use can cause ice formation inside the pipe unless the compressed air supply is dosed or dry.

Outlet ports are a particular source of ice formation because of the sudden drop in air temperature at this expansion point. Ice can build up quickly, blocking the port. Dosed or dry air cannot guarantee protection from icing because the port is exposed to outside air. If protection from exhaust port icing proves troublesome then two possible solutions are:—

(i) apply alcohol dosage at or immediately preceding the port (mostly applicable to a continuously operated system);
(ii) apply localized heating to the port.

High Pressure Pneumatics

FROM AN overall efficiency point of view, optimum usage of compressed air is restricted to compression ratios of 7:1 to 8:1 (*ie* working pressures of 7 to 8 bar). The development of higher working pressures requires an increasingly disproportionate expenditure of power for compression, as well as increased component stresses and explosion-hazard in the event of failure. The marked difference between pneumatics and hydraulics in these respects is due to the compressible nature or 'elasticity' of air.

For general applications, the only advantage offered by higher compression ratios is in storage systems where high pressure air can be stored in a suitable pressure vessel and 'let down' to conventional delivery pressures. Storage at high pressure substantially reduces the volume required for the storage vessel.

For other more specialized applications there are definite requirements for higher pressure compressed air, which may extend to pressures of around 1 000 bar for air blasting and metal forming – see also Table I. High pressure air systems can, in fact, be categorized in four pressure ranges, with conventional industrial pneumatics rated as low pressure systems.

(i) *Moderate pressure (17.5–35 bar),* normally derived directly from a compressor, or reservoir charged by a compressor.

(ii) *High pressure (35–210 bar),* normally derived from a charged air bottle.

(iii) *Very high pressure (210–450 bar),* either derived from high pressure reservoirs charged by multi-stage compressors for large volume applications, or from intensifiers for 'single slot' or small volume application.

(iv) *Ultra-high pressure (700–1 400 bar),* for highly specialized applications with supply from multi-stage reciprocating compressors.

The direct production of high pressures demands the use of multi-stage compressors of the reciprocating type. Because output is necessarily restricted, a compressor (or batch of compressors) can be used to charge up a reservoir to the required pressure, the reservoir volume being selected on the basis of the working capacity required. Alternatively, if the demand is relatively low, it can probably be met by a gas bottle where the initial pressure available is usually of the order of 125 bar (1 800 lb/in^2). If the working pressure required is very much lower, this again may feed a reservoir, through a pressure reducing valve. The third alternative is to obtain the high

TABLE I – HIGH PRESSURE PNEUMATICS

Category	Pressure Range bar	Typical Applications	Working Pressures bar	Working Pressures lb/in^2
Moderate	17.5–35	(i) Engine starting	25–42	350–600
		(ii) Air blast switchgear	17.5 up	250 up
		(iii) Deep sea diving	10	150
High	35–210	(i) Sea diving (air cylinder charge)	up to 210	up to 3 000
		(ii) Deep sea diving (air cylinder charge)	105–140	1 500–2 000
		(iii) Secondary/emergency aircraft control services	35–210	500 3 000
Very high	210–450	(i) For specialized process plant	210–350	3 000–5 000
		(ii) Supersonic wind tunnels	210–420	3 000–6 000
Ultra high	700–1 400	(i) Air-blasting	700–840	10 000–12 000
		(ii) Metal forming	up to 14 000	up to 20 000

TABLE II – THEORETICAL POWER TO COMPRESS 1 CUBIC METRE OF AIR AT ATMOSPHERIC PRESSURE

Power in Kilowatts

	Delivery Pressure (bar)					
	7	70	140	210	280	350
Two-stage	–	10.0	–	–	–	–
Three stage	–	8.9	10.7	11.8	–	–
Four-stage	–	8.4	10.4	11.0	11.8	12.4
Five-stage	–	–	–	10.5	11.2	11.8
Isothermal compression	3.6	7.1	8.3	9.1	9.5	9.7

TABLE IIA – THEORETICAL POWER TO BOOST 1 CUBIC METRE OF FREE AIR PER MINUTE

Power in Kilowatts

Inlet Pressure	Boost	Delivery Pressure – bar				
		70	140	210	280	350
7	Multi-stage	4.5	6.0	6.70	7.25	7.82
70	Single-stage	–	1.3	2.20	3.00	3.35
140	Single-stage	–	–	0.75	1.30	1.86
210	Single-stage	–	–	–	0.56	1.00

pressure supply indirectly by boosting an available low pressure supply. This may be a normal air line supply, *eg* 7 bar (100 lb/in^2), or a supply from a gas bottle (say at nominal 105 bar (1 500 lb/in^2)) according to the final pressure required and the pressure intensification possible from the booster. (See also Tables II and IIA).

All three methods tend to have the common limitation that the volume of high pressure air available is restricted, either in terms of free air equivalent, or in the time over which the required working pressure is available. The higher the working pressure level, the more marked this restriction becomes. Thus high pressure pneumatics tends to be limited to specialized systems or requirements which only need short term or non-continuous operation.

This convention may, however, be subject to change. In certain fields, the use of higher pressures under exact control may provide the higher output forces normally associated with hydraulic power, particularly for 'impulse' applications, and in servo mechanism working. Even the methods of obtaining air pressure may change, such as by using liquid air for storage, or even generation of pressurized air by chemical processes.

Currently, practically all the compressed air (and other high pressure gases) for industrial use are derived from conventional multi-stage compressors. Only the lower pressures are normally supplied direct from compressors. For higher pressures up to about 140 bar (2 000 lb/in^2) the most convenient form of supply is usually a charged gas bottle. For pressures of 105 bar (1 500 lb/in^2) up to about 1 400 bar (20 000 lb/in^2) a gas booster is used.

A *gas booster* is simply a pressure intensifier or air pump accepting low pressure air in a low pressure cylinder, to drive a smaller piston in a high pressure cylinder. This tandem configuration is associated with a valve mechanism which acts at the end of each stroke to produce continuous reciprocation. The high pressure piston and cylinder form the upper portion of the intensifier or pump and the inlet and outlet valves are normally mounted on the end of this cylinder where they are readily accessible.

The inlet is compressed air at normal line pressures (eg 7 bar or 100 lb/in^2 gauge). The outlet pressure is determined by the ratio of the areas of the low and high pressure pistons offering, in theory at least, a direct method of boosting the supply to almost any pressure required. In practice, however, performance is limited by the unswept volume present (which is kept to a practical minimum), and that optimum performance is achieved with a pressure ratio of outlet pressure to inlet pressure between 4:1 and 6:1.

To obtain higher outlet pressures with economic and efficient booster working, a charged gas bottle can be used to provide the initial supply. For still higher pressures two boosters can be used, operating in series. To reduce pressure fluctuations in the supply to the second booster a receiver can be incorporated between the two.

Basically, a booster continues to pump until a preset pressure is reached. It will then continue to deliver air at this higher pressure, or will remain stalled with no demand, until the pressure falls below the preset level. Since the output pressure is directly related to the input pressure, any variations in the latter will affect the output pressure. Thus, when the supply is from a gas bottle, pressure from the supply will fall as the bottle is discharged. The maximum pressure available from the booster is then based on the lowest pressure to which it is considered economic to let the bottle pressure fall. This may be further modified by the output volume required.

Let-Down Systems

Compressed air stored in bottles or high pressure receivers provides a cheap, compact source of emergency power. It can also be used for portable systems, divorcing the actuator or appliance

from a compressor. The supply pressure required may be low – eg a conventional 7 bar (100 lb/in^2) supply – when it can be advantageous to store the compressed air at a much higher pressure in order to increase the volume of free air available. The storage vessel can then be fitted with a reducing valve to supply air at the required pressure. A shut-off valve necessary to isolate the pressurized supply vessel, cannot provide satisfactory pressure control. If the pressure level to be maintained is critical, then further control may be provided by a back-pressure or pressure-maintaining valve.

A *back-pressure valve* is shown in Fig 1. This is of the 'inverted' type, with inlet pressure tending to close the valve rather than open it. The valve is opened by inlet pressure lifting a spring-loaded piston to which the poppet is loosely connected. The poppet valve is seated by a light spring, the inlet pressure acting on the small sealed area. Before the valve can open, the spring-loaded plunger must lift sufficiently to take up the slack between it and the poppet.

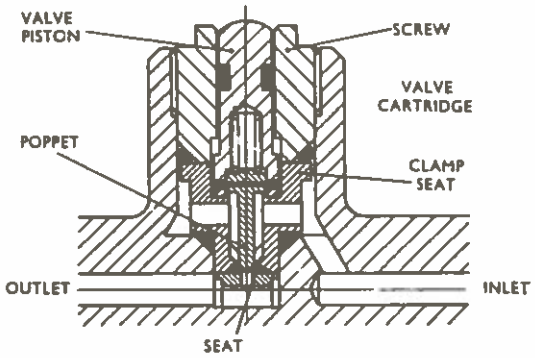


Fig 1

Unlike a liquid, high pressure air (or gas) is highly compressed and, on release of restraint, will tend to revert instantaneously to its original free air volume, within limits imposed by the nature of compressible gas flow. The very high compression ratios which may be present (see Table III)

TABLE III – COMPRESSION RATIOS

Final Pressure		Compression Ratio
bar	lb/in^2	
7	100	7.8
14	200	14.6
21	300	21.4
28	400	28.2
35	500	35.0
70	1 000	69.0
140	2 000	136.5
210	3 000	204.6
280	4 000	273.0
350	5 000	340.0
700	10 000	680.0
1 055	15 000	1 020.0
1 406	20 000	1 364.0

indicate the very high energy which may be stored in even a small volume of high pressure air, and the potential explosive hazard if suddenly released. Special care and precautions are, therefore, necessary with all high pressure systems and appliances.

Conventional copper piping is generally unsuitable, and nylon tubing quite unsuitable, for high pressure lines. All pipework should be of hydraulic quality steel, or preferably stainless steel, for pressures in excess of 350 bar (5 000 lb/in^2). Fittings and couplings must be of compatible design and rated for the pressures involved for use with air (gas pressure ratings differ from hydraulic pressure ratings, with most types of compression couplings).

Pressure vessels should be the Safety Gas Pressure type, and free from oil. No pressure vessels, eg receivers and gas bottles, should be used unless they have a current test certificate. By law, all receivers and gas bottles must be insured, and the insurance company is responsible for carrying out the necessary certification tests.

Oil in any form is not tolerated in a high pressure gas system since its presence can cause 'dieseling' when the gas is introduced.

Engine Starters

Compressed air motors used for engine starting are normally powered from a receiver charged to conventional low air pressures. Much higher air pressures are, however, used on starters for marine engines, when the receiver pressure may be of the order of 25–35 bar (300–500 lb/in^2). The receiver in this case is charged directly by a compressor either from the main engine power take-off, or from its own electric motor. A reducing valve is used between the receiver and the starter motor or engine, as necessary.

Two distinct methods of using the compressed air are, in fact, utilized. In the one case, the air drives a conventional air motor or rotary actuator working as a starter motor. In the other, compressed air may be directly injected into a selected number of engine cylinders to achieve rotation for starting. In this case higher air pressures may be used directly.

Air Blasting

The 'explosive' nature of highly compressed air is used in air blasters or airbreakers, this technique having replaced the use of conventional explosives in coal mines to some extent. The working principle involved is that air at very high pressure (eg circa 850 bar [1 200 lb/in^2]) is suddenly released in a conventional bored shot hole, the resulting release of stored energy producing first, rupture of the coal in the locality, and then further break-up by the expansion of the escaping air into the fissures so formed. The action is not so violent as that produced by explosives, but is generally more far reaching. Thus, a greater proportion of coal is dislodged per blast. The amount of coal dislodged depends primarily on the thickness of the seam and can vary from three to fifteen tons per shot.

High pressure air is introduced into the borehole via a special breaker assembly – eg Fig 2. Basically, this consists of a steel shell, sealed at one end by a shear disc inserted between two shear rings and retained by a shell head. The shell head also contains the backward facing discharge ports. A supply of high pressure air is connected via a reinforced hose to a blasting valve. When this valve is opened the pressure in the shell rises until the shear disc ruptures. Air is then free to expand through the discharge ports into the bore hole, producing a jet force causing fracture of the coal. At the same time, the fact that the ports are backward facing drives the shell further into the hold, increasing the effectiveness of the blast.

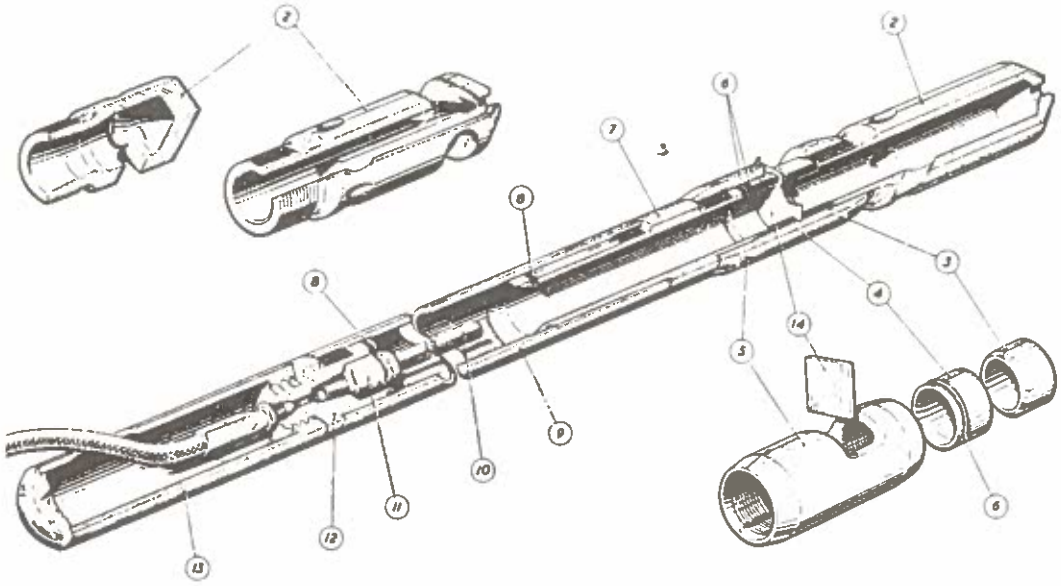


Fig 2 Bursting shell for air blasting coal. 1. 2½ in Breaker assembly. 2. Retainer head. 3. Shear ring. 4. Seal ring. 5. Body. 6. O-ring. 7. Adaptor. 8. O-ring. 9. Tube. 10. Cooling tube. 11. Inlet fitting. 12. End cap. 13. End guard. 14. Slugs.
(Olin Mathieson Ltd).

Air blasters of this type used in mining are generally supplied by multi-stage reciprocating compressors, located at some convenient central point. Hose lengths may run to many miles. Similar types of air breakers have also been used for blasting concrete and other materials. The rupture action is, however, somewhat different in materials having higher ruputure strengths and different fracture characteristics.

Wind Tunnels

Wind tunnels, designed for working with airstreams of high Mach number, are normally of the intermittent type. That is, the high flow velocities required are achieved by a sudden discharge of high pressure air from a reservoir. Reservoir storage pressure may be of the order of 210–420 bar (3 000–4 000 lb/in^2). Multi-stage compressors are used to produce this charge, the number and capacity determined by the charge-up time required. Commonly it may take the compressors several hours to charge the reservoir to a pressure level sufficient for a single test lasting a few seconds.

Whilst a relatively simple method of obtaining high air velocities, a basic problem with this application of high pressure air is the necessity for the air to be completely dry. The sudden expansion of the compressed air causes a severe drop in temperature, which would mean that any moisture present in the air would immediately be turned into ice particles, projected through the tunnel at high velocities. For similar reasons it is also desirable to remove all traces of oil vapour and any solid particles from the air.

High Energy Rate Forming

High pressure air has been used successfully for high energy rate forming of sheet metals, the main requirement being that for maximum effectiveness the high pressure air should be applied instantaneously. Again, this is commonly applied through a bursting disc, although it may be further controlled by a quick-opening valve. The method is essentially similar to explosive forming but avoids the use (and storage) of potentially unstable materials.

Air Blast Switchgear

Compressed air at a pressure of 17 bar (250 lb/in^2) upwards is employed on modern switchgear for cooling and extinguishing the arc formed by opening contacts breaking a high voltage circuit. The resulting improvement in contact life can be substantial.

Aquasports and Skin Diving

Air supply for underwater swimming is provided by lightweight air bottles, charged with air pressures up to 210 bar (3 000 lb/in^2). The purity of the air is very important in such cases. It must be oil free and clean. If the compressor used for charging the bottles is not an oil-free type, it must be followed by an oil separator. The air is also filtered before filling the gas bottle, the filter often being charged with alumina, or silica gel, and activated charcoal.

The total quantity of air contained in a charged bottle will be dependent on the temperature, as well as the pressure, and so it is an advantage to deliver the air to the cylinder at as low a temperature as possible. Rapid filling of the bottle can cause a rise in temperature, and in such cases further topping up may be possible when the bottle has cooled down.

Vacuum Techniques

VACUUM TECHNOLOGY embraces all systems utilizing a pressure less than that of the atmosphere, normal atmospheric pressure being taken as 1.013 bar. The complete vacuum range down to the ideal vacuum is split into five classes, *viz*

(i) *Low vacuum* — pressures down to 25 torr

(ii) *Medium vacuum* — pressure range 25 torr down to 10^{-3} torr

(iii) *High vacuum* — from 10^{-3} down to 10^{-6} torr

(iv) *Very high vacuum* — from 10^{-6} down to 10^{-10} torr

(v) *Ultra high vacuum* — less than 10^{-10} torr.

(See also Fig 1).

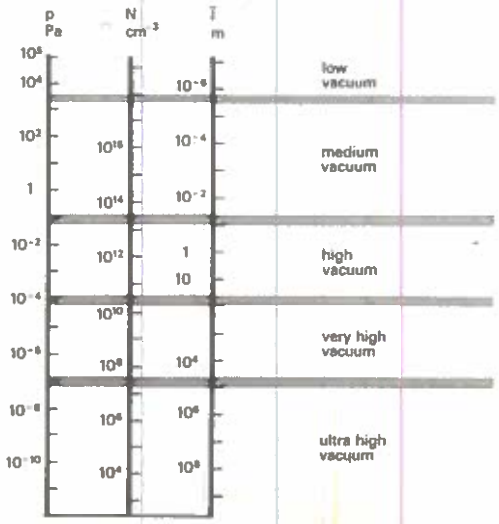


Fig 1 Classes of vacua.

VACUUM UP TO 97%

with maximum reliability at the minimum cost

Before designing anything which involves a vacuum pump or compressor it makes sense to get the benefit of the best advice available.

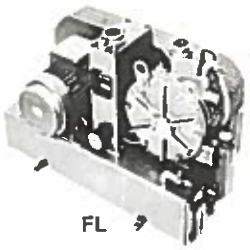
By involving Lacy-Hulbert right at the beginning, you make sure you're getting exactly the right equipment for your application. That way, you not only get the maximum reliability – you also get the minimum cost. (Don't forget, a too-small pump can cost you just as much with its inefficiency as a too-big one with its over-specification for the job.)

Lacy-Hulbert have been engineering vacuum pumps and compressors for over 75 years. And during that time they've met – and solved – just about every problem known to the pneumatics business.

Shown here are just two items from Lacy-Hulbert's wide range of equipment.

Lacy-Hulbert R Series
The R Series consists of a range of rotary sliding-vane type air-compressors and vacuum pumps. Maximum operating pressures can be up to 1 bar or a vacuum of 80% depending on the model selected. Displacements range from 1.5 m^3/HR to 88 m^3/HR. Oil-free or drip-lubricated versions are available.

Lacy-Hulbert FL Range
This is a range of rotary sliding-vane type vacuum pumps which are flood-lubricated for high performance. Displacements range from 17 m^3/HR up to 40 Bm^3/HR, and the maximum vacuum is 97%. The FL pumps are designed as self-contained vacuum sources with extremely low noise-levels.


FL

UK Sales and Service
The UK is covered by a comprehensive organisation of regional sales engineers and agents.

A team of service engineers provides a nationwide coverage for in-plant servicing and, if required, regular routine maintenance visits can be made under our Contract Service Scheme.

International sales and servicing are handled by Lacy-Hulbert agents in well over 30 countries, including Europe, Asia, Australia and the USA.

For full details of all Lacy-Hulbert equipment and facilities, or for expert advice on any type of installation, contact Lacy-Hulbert at the address below.

Lacy-Hulbert & Company Ltd
Richmond Road, Beddington
Croydon CR9 4EA
Telephone: 01-680 5226
Telex: 21598

THE BEST PERIODICAL DEVOTED TO NOISE & VIBRATION CONTROL

Packed with practical, up-to-date, worldwide news, information and data,
NOISE & VIBRATION CONTROL Worldwide covers all aspects of this important subject.

This journal deals with its causes, effects, measurement, and methods of control. As legislation in all countries becomes more stringent, so too does the demand for this detailed specialist journal.

TRADE AND TECHNICAL PRESS LTD.,
CROWN HOUSE, MORDEN, SURREY, SM4 5EW, ENGLAND.

The most common use of vacuum in industrial processes is to reduce the boiling point of water contained in a product in order to achieve one or more of the following ends:

(i) to prevent thermal damage to the product when heat is applied to vaporize and to effect mass transfer of the water and thus dry the product,

(ii) to speed up the process, and

(iii) to ensure very low residual moisture content in products difficult to dry by other means. Examples of this application are distillation, evaporation, vacuum drying of liquids and solids and freeze drying processes.

Vacuum cooling is closely allied to vacuum drying and is a simple mass transfer process which results in cooling of the product by the removal of a controlled quantity of moisture.

Depression of the vaporization point is also the objective of vacuum coating processes in which metals and other materials are melted, evaporated and eventually sublimated on prepared substrates. The high vacuum condition prevents oxidation of the material evaporated at high temperature.

Simple removal of atmospheric air and of moisture so that they can be replaced by another gas or liquid is the reason for the use of vacuum in impregnation; in filling of capillaries and containers with small orifices; in some vacuum investment and die casting processes and in refrigeration, cryogenic and gas filled cooling system evacuation; electric lamp and switch gear evacuation and inert gas welding systems.

In vacuum packaging of food, vacuum is applied to remove atmospheric air and its oxygen from the pack so that the contained product will have an increased shelf life, once it is sealed with an impervious plastic pack. An important aspect of vacuum packaging for some products is that the differential pressure between the exterior and the interior causes the flexible pack material to collapse and cling to the product, providing increased structual strength.

'Mechanical' Processes

Almost as numerous are the various ways in which a low vacuum can be used for mechanical handling and similar low-cost automation. Some examples are:-

High speed paper feeding for printing, processing, etc.

Vacuum forming of short thermoplastics

Liquid filling

Dry powder filling

Continuous liquid/solid separation

Mobile materials handling

Air sampling

Printing

Vacuum feeding sheet steel, glass, etc.

Machine tool chucks and vices

Miscellaneous applications include:

Automatic addressing, Assembling, Book binding, Shoe making, Bread wrapping, Bottle inspection, Carton opening and feeding, Cigar banding, Collating, Etching,

Paper folding, Automatic grinding and polishing, Labelling, Money counting and banding, Automatic bread handling, Papermill sheeting, Photo-copying, Ruling, Tag handling, Tile cutting, Perforating, X-Y plotting, Punch press feeding, Check sorting, Automatic welding, *etc, etc.*

Selection of Pumps (see also Fig 2)

The production of vacuum pressure implies exhaustion of a given vessel by suitable pumping action. Certain types of compressors can also work as exhausters, but in general the machines used for producing vacuum pressures are known as pumps (vacuum pumps).

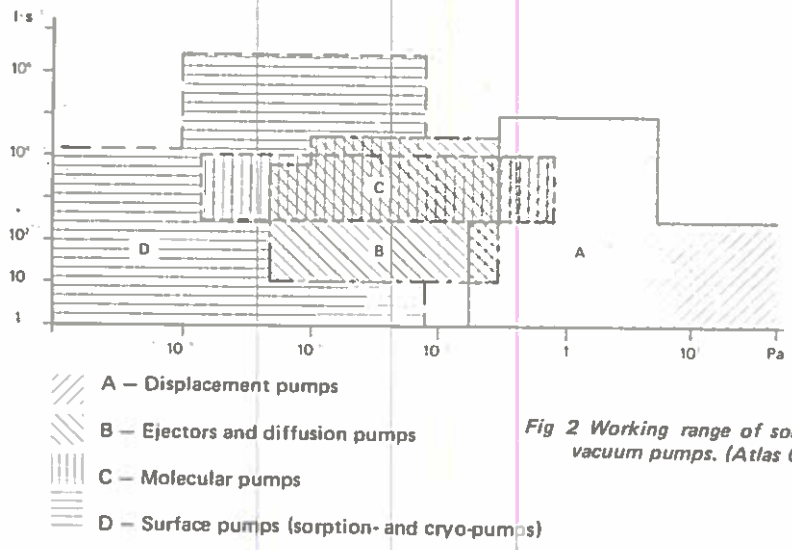
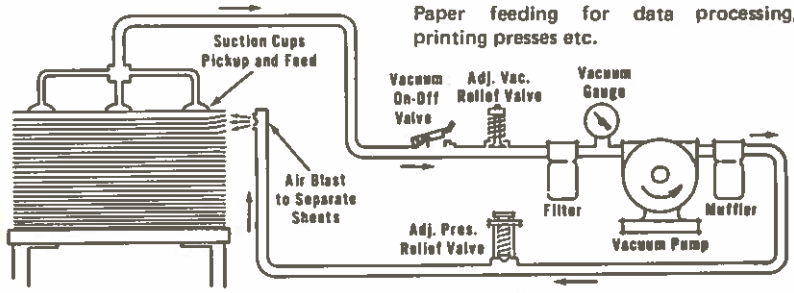


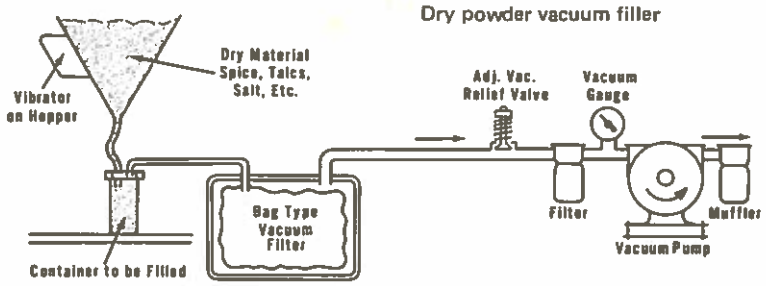
Fig 2 Working range of some types of vacuum pumps. (Atlas Copco).

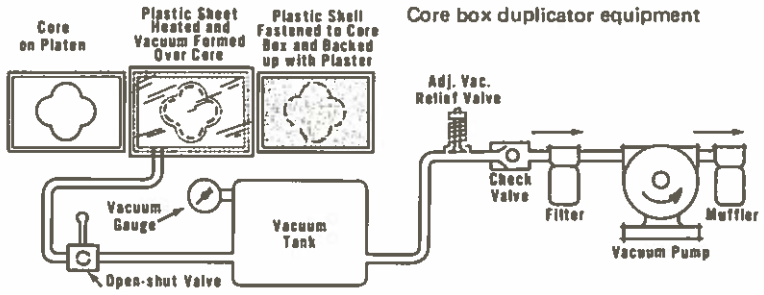
The production of vacuum pressure implies exhaustion of a given vessel by suitable pumping action. Certain types of compressors can also work as exhausters, but in general the machines used for producing vacuum pressures are known as pumps (vacuum pumps).

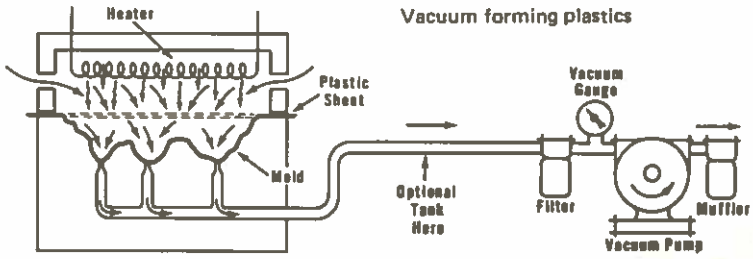
The pump or pump combination is selected on the capacity needed for the range to be covered, with due consideration to pump down time and duration of cycle, the latter factors determining the most suitable pump size. The latter is usually a straightforward matter in the case of coarse vacuums, for the pump size is related solely to the size of the container. In the medium and medium-high vacuum range the shape as well as the volume of the container must be taken into account, the lower the final pressure, the greater the proportionate effect of the surface area factor. Selection of pump capacity in this range tends to become somewhat complicated and is normally best based on empirical data.

In the high and ultra-high vacuum range the pump size required is related almost entirely to the size and state of the surface of the container. Thus at pressures below 10^{-3} torr there are more adsorbed or absorbed molecules in the system than free gas molecules. The performance of the pump must thus ensure that at the required working pressure a rate of evacuation is maintained equivalent to the rate at which adsorbed and absorbed gas is emitted from the surfaces of the system. This can vary widely with different surfaces.



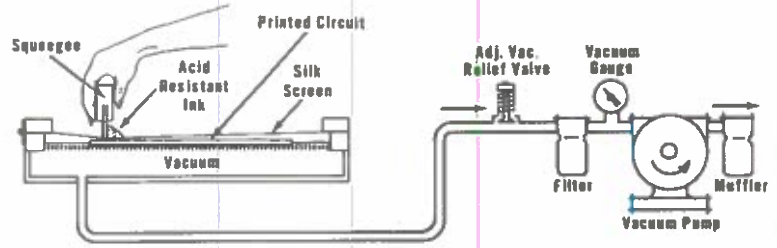




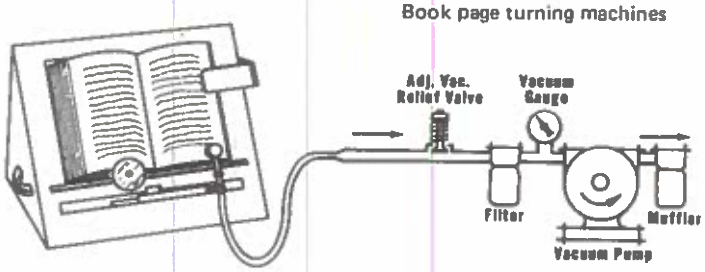


Examples of low vacuum application
(Gast Manufacturing Co Ltd)

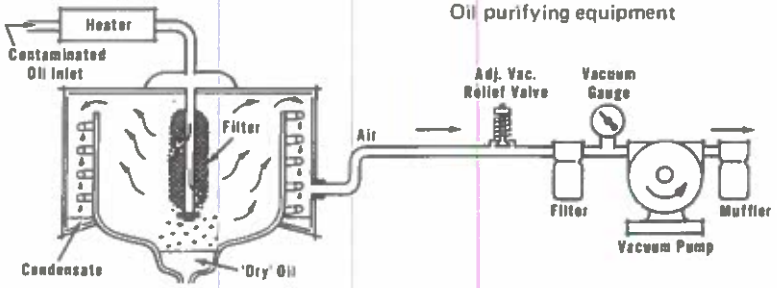
Silk screening printed circuits



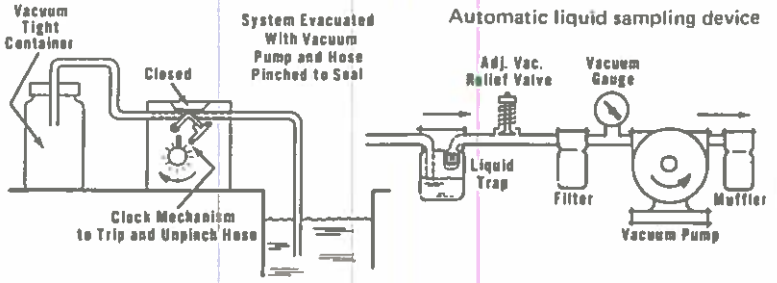
Book page turning machines



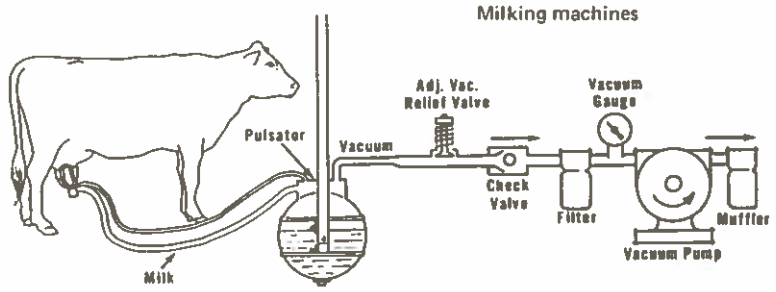
Oil purifying equipment



Automatic liquid sampling device



Examples of low vacuum application
(Gast Manufacturing Co Ltd)



Example of low vacuum application (Gast Manufacturing Co Ltd)

This condition may be further complicated in practical systems by the fact that processes take place within the container being evacuated, whereby vapours are given off — eg in drying, degassing, distilling, etc. In such cases it is necessary to form an accurate estimate of the amount of vapour likely to be released or accumulated during the process so that the pump can be sized accordingly, bearing in mind also the nature of the vapour. Where large quantities of vapour have to be dealt with it is often necessary to install an efficient condenser system before the pump.

The working pressure to be produced or maintained in the system thus determines the type of pump required (or pump combination, where a fore-pump is necessary), and also indicates whether the pump size is dependent primarily on volume or total vacuum surface area. Where additional gases or vapours are present as a result of processing, their volume and properties also have to be taken into account. Finally the pumping down time has to be considered in arriving at a suitable pump size, balanced against the operating characteristics involved — eg whether the system exhausts as rapidly as possible down to the working pressure as part of definite work cycles, or whether the process is a continuous one with the pumps continuing to operate after pumping down to the working pressure.

See also chapter on *Vacuum Pumps.*

Air Films

THE USE of an air cushion to reduce friction to a minimal level is well known in applications ranging from the floating of one accurately machined or otherwise smooth surface over another to the suspension of heavy loads.

The basic form of an air suspension unit is shown in Fig 1, together with the corresponding pressure distribution diagram.

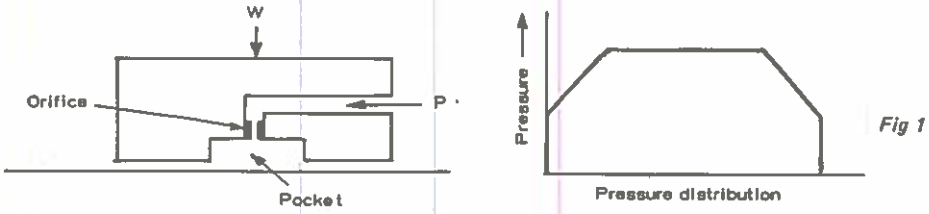


Fig 1

Basically the load bearing capacity of a simple air-lift pad is given by:

$$W = K.P.A.$$

where

W = bearing load

P = air supply pressure

A = bearing area

K is a constant depending mainly on the orifice or orifice pocket dimensions. This may approach 0.25 for a simple orifice, but can be increased by suitable modification of the orifice pocket and depth controlling the thickness of the air film.

A suitable design and arrangement of air bearing pads can thus provide a complete low-friction transport system, either incorporated in the design of the subject (as in hovercraft), or as a fixed system over which the subject travels (as in an air slide or air-lift conveyor).

The air cushion concept developed for vehicle suspension is relatively complex in terms of the pneumatics involved since it also involves stability and control as well as generation of direct lift.

Significantly, however, the maximum contact pressure developed is very low (*eg* generally less than 0.2 bar (4 lb/in^2)) or considerably less than that of a man's footprint (typically 1 bar). Thus a large cushion area is needed to suspend the total weight of the vehicle. Also the limit of stability is set by two frequencies — the natural frequency corresponding to the static stiffness of the system (which depends only on the leakage gap or 'pendulum' frequency); and the pseudo acoustic frequency (which is related to the ratio of the mass of air to sustained vehicle mass). Analysis of such a system becomes extremely complex, so much so that much of the original design work on hovercraft was done on purely empirical lines.

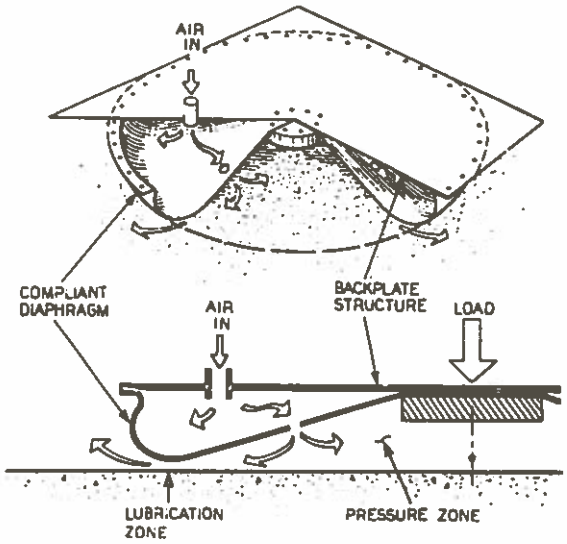


Fig 2 Structural fluid bearing.

Lesser known is the compliant air lift bearing for bearing transport, normally designed to work off on conventional compressed air supply of 5—6 bar. This is based on a compliant inflatable diaphragm which, on being inflated, conforms to the shape of the surface it is to support and lifts the load until a controlled air leakage to atmosphere forms a thin air film between the diaphragm and the surface — Fig 2.

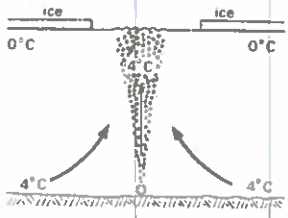
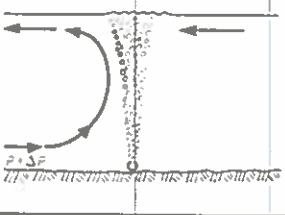
Basically, as the diaphragm inflates lift is created and the load rises slightly. The air then flows through holes inside the diaphragm, filling the cavity beneath until diaphragm area multiplied by air pressure equals the weight of the load. Air then escapes between the diaphragm and the surface establishing the thin film of air that enables virtually frictionless movement to take place. Because the load is fixed, and the diaphragm is compliant, the air gap remains a constant all the way round its periphery. The only friction involved in moving the load is between the molecules of air in the air film itself.

Air Bubble Techniques

AIR BUBBLE techniques are based on the 'bubble barrier' produced when compressed air is fed into a submerged, perforated hose creating a series of bubble plumes rising from the holes. The rising bubbles cause a vertical current of air and water to flow to the surface, in turn generating a flow of water towards the barrier in the lower layer of water; and flow away from the barrier in the upper layer.

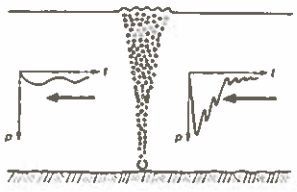
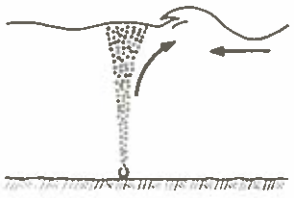
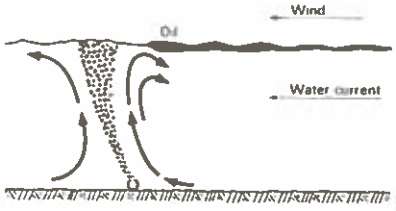
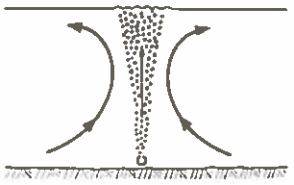
The table describes and illustrates various techniques based on this principle and developed by Atlas Copco.

AIR BUBBLE TECHNIQUES

Application	Diagrammatic	Description
Ice-prevention		The bubble barrier in this case transports warmer bottom water to the surface, creating an ice-free area along the barrier. It has been used successfully to prevent ice damage in yacht marinas.
Reduction of salt intrusion		Here the bubble barrier stops and reverses the intrusion of salt water into flowing fresh water, much of which is then carried back out to sea by the fresh water stream. This scheme is used in several Dutch locks.

cont...

AIR BUBBLE TECHNIQUES (contd).....

Application	Diagrammatic	Description
Underwater blasting		The bubble barrier can considerably reduce the effect of shock waves when underwater blasting by reducing the maximum amplitude of the shock wave, *ie* subduing peak pressures. It cannot, however, reduce the total energy of the shock wave.
Pneumatic breakwater		Provided the generated surface velocity produced by the bubble barrier is high enough — *ie* exceeds 25% of the propagation velocity of the waves — then steep waves will break or be reduced in height. High airflow rates are required. This system has been used in Japan.
Oil-protection barriers		A bubble barrier can prevent the spread of oil slick from an oil spill by the surface velocity generated by the barrier. This system has been shown to work well in practice, but is still subject to further development.
Lake restoration		Basically, in this case, an air bubble generator is used to treat polluted lakes by aeration; also to promote water circulation in deep lakes. Primarily it can offer a solution where the main problem is oxygen deficiency.

Pneumatic Conveyors

PNEUMATIC CONVEYANCE of solids via pipelines can be broadly classified by the operating pressures involved:

(i) Low pressure systems
(ii) Medium pressure systems
(iii) High pressure systems

Equally, successful transport of solids is directly related to air velocity: a certain minimum velocity is necessary to maintain solids in suspension, or in a fluidized state, in horizontal pipelines; and on even higher minimum velocity to sustain transport without choking in vertical transport.

Low Pressure Systems

In low pressure systems, operating pressures may be as low as 1—2 inch water gauge to a maximum of about 0.3 bar differential. They are thus largely restricted to the transport of light, bulky materials with relatively large surface area.

The differential pressure may be either negative or positive — *ie* conveying produced by suction or blowing, using either an exhauster or fan, respectively (although in some cases the two methods are combined with the material being conveyed pulled directly from one side to the other of the fan). The suction method is generally cheaper to install than a blowing system and is also particularly adapted to systems where there are to be several pick-up points and one delivery point. In this case the pick-up can usually be by hand nozzles. The other basic advantage of the vacuum method is that although the concentration ratio with low pressure systems tends to be low, suction systems achieve a higher concentration through smaller bore pipes than a blower system operating att the same pressure differential and thus reduce the amount of air needed for conveying.

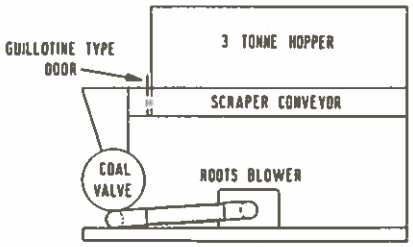
The chief disadvantage of suction systems is that with a low pressure differential capacity is limited and the maximum practical length of pipeline is also limited. This is a limitation of all low pressure systems, but with blowing the pressure can be increased, if necessary, to achieve higher concentrations. Thus blowing systems are usually preferred when the handling rate exceeds 5 tons per hour, although this also depends upon the type of material being handled and the conveying

requirements. Suction pick-up is an important feature where there are several pick-up points in the system. Blowing systems tend to become most economic when there is only one pick-up point and several discharge points.

Maximum suction pressure normally realizable with vacuum systems is of the order of 0.3 bar (5 lb/in^2) differential, at which level the system is classified as a medium pressure type. The corresponding maximum differential for medium pressure blowing systems is of the order of 1 bar (15 lb/in^2). Medium pressure suction systems are widely used as unloaders from bulk conveyors, such as wheat, maize and other similar granular materials. Since such permanently installed systems often operate through a considerable vertical lift via swivelling booms they are sometimes termed pneumatic elevators. The vertical pipelines may also accommodate telescopic sections to suit the rise and fall of the bulk conveyor, *eg* cargo ships or barges.

Medium Pressure Systems

Medium pressure blower systems normally derive their differential pressure from a blower or compressor (as opposed to a fan, which may be used for a low pressure system), and are sometimes described as dense stream conveyors because of their ability to handle heavy material loads. Such loads include a wide variety of powdered, chipped, granular and pelletized materials and the material is invariably conveyed in a 'fluidized' state. The quantity of air, and the pressure differential required, is dependent on a variety of factors including particle size and shape, bulk density, moisture content and the aerodynamic characteristics of the system. Satisfactory solutions are normally determined on empirical lines, with power requirements and capacity specifically related to the material being handled.



Delivery unit for pneumatic transport of coal. (National Coal Board).

High Pressure Systems

High pressure systems are invariably of the blower type and work at a pressure differential in excess of 1 bar. They are used for the handling of heavier products, including ores, cement, clay, bauxite, silica, *etc*; and even small metal components and similar solid objects. Apart from the fact that a high pressure blower is used as the power source (normally a centrifugal blower), the main difference between a high pressure system and other systems is normally the method of feeding the material into the system. It is necessary to maintain an adequate pressure seal during feeding, this usually being accomplished by means of a blow tank when the pressure differential involved is 3 bar or greater.

Basically, a blow tank comprises a sealed vessel into which the material is fed under gravity via an inlet valve which is closed at a predetermined level of filling to render the vessel air tight. Compressed air is fed to the vessel, *via* a fluidizing ring or fluidizing chambers or pads, the fluidized material being discharged into the conveyor system through an outlet at a predetermined pressure level. Outlets are fitted either to the top or the bottom of the pressure vessel, depending on the

characteristics of the material to be conveyed. Feed is accomplished on a cycling basis, *ie* is intermittent, but where effectively continuous action is required two blow tanks may be employed operating on an alternating cycle.

Feed and Air Losses

For low pressure systems continuous feed can be provided by suction nozzles in the case of suction systems, or from a hopper in the case of blower systems. The hopper may provide uncontrolled entry of the material, or entry may be controlled by means of a trickle valve or a simple rotary valve.

With medium pressure systems air losses can be minimized by the use of an air lock in the feeder system. This can be based on two simple flap valves or two rotary valves with an intermediate chamber. This provides a semi-continuous feed although the sequence of emptying and filling of the air lock can be repeated quite rapidly to give the overall effect of a continuous feed. Besides sealing the entry against air losses, semi-continuous feeders of this type also reduce abrasive problems when the material being handled is abrasive by nature.

Except for closed circuit systems, filters are usually an essential part of any pneumatic conveyor system. Since the material being handled is conveyed in a 'fluidized' state, separation of the conveying air and product is achieved at the delivery point when dust, *etc*, will tend to escape with the air. It may be desirable or even obligatory to reduce the amount of dust discharged to the atmosphere to a minimum; and in some cases necessary as an economic feature to recover such solids as may remain in the discharged air.

Filters used range from filter bags or sleeves to wet filters, water curtains, cyclone separators and electrostatic precipitators. The type chosen is normally directly related to the product being handled and the performance required.

Related Devices

Related devices, relying on fluidization or suspension are summarized in Table I.

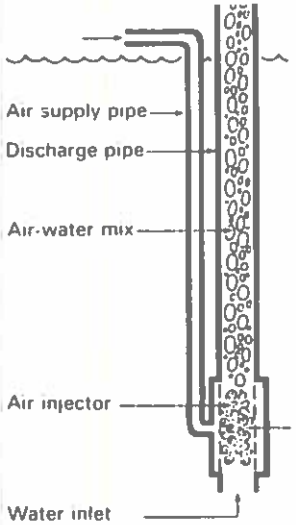


Fig 1

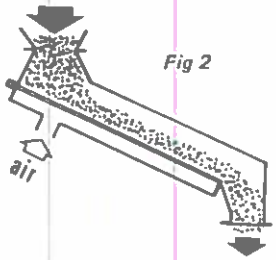


Fig 2

Fig 3

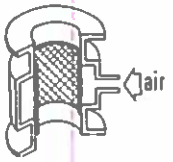


Fig 4

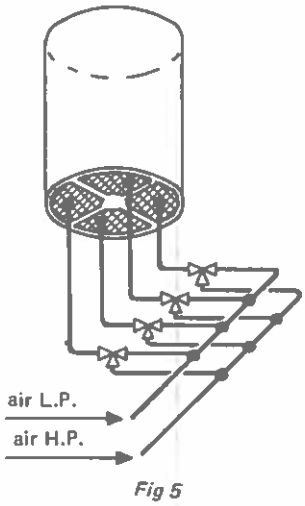


Fig 5

TABLE I – PNEUMATIC HANDLING SYSTEMS

Name	Diagrammatic	Principle
Air-lift pump	Fig 1	Injected air mixed with water at the bottom of a vertical pipe produces a lower density mixture which can be lifted to a considerable height. Solid materials less than the diameter of the pipe can also be lifted.
Air slide	Fig 2	Air-actuated gravity conveyor, low pressure air applied through a porous bottom of the slide lifting and suspending the product.
Pneumatic elevator	Fig 3	Compressed air fed into a vertical chamber through a fluidizing ring 'floats' the solids and lifts them to the top of the column. Particularly suitable for handling delicate products.
Fluidizer	Fig 4	Device for aerating a solid product thus reducing its apparent density by fluidization. At a specific air velocity the product will behave as, and can be handled like, a fluid.
Blender	Fig 5	Silo fitted with separate aeration pads connect to both low pressure and high pressure air supplies. The low pressure supplies flotation of the product. Alternate application of high pressure air to the aeration products tumbles and blends the contents of the silo.

Compressed Air for Breathing

MAXIMUM ACCEPTABLE levels for impurities in breathing air supplied by compressed air systems (as given in BS4275:1974) are:-

carbon dioxide 500 ppm (900 mg/m^3)
carbon monoxide 5 ppm (5.5 mg/m^3)
oil mist 0.5 mg/m^3

Breathing air must also be free from odours and contamination by dust or metallic particles and any other irritating or toxic contaminants.

Of these, only oil mist is likely to be present in significant concentrations in air delivered from oil-lubricated compressors, provided the compressor itself can induct only clean, uncontaminated air. Carbon dioxide and carbon monoxide should normally be absent provided the compressor is adequately sized so that it does not overheat.

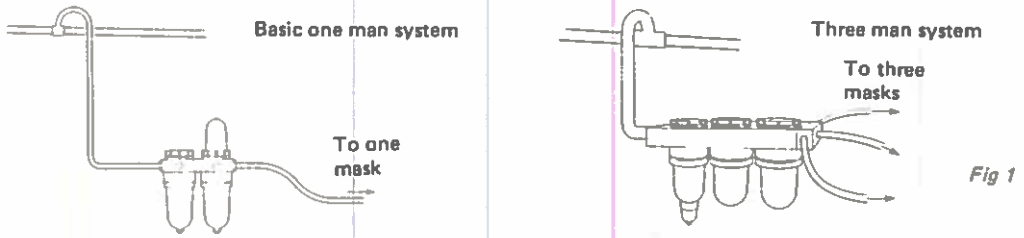


Fig 1

In industrial systems the breathing air supply is commonly tapped directly from the main airline (Fig 1). This places a premium on correct airline installation, even though the breathing air is necessarily separately filtered. Where there are multiple working points demanding breathing air supplies it is preferable, however, to install an airline for breathing systems separate from the normal airline or working supply line. (Fig 2).

A combination of filters is required following the take-off point for each breathing airline. The first can be a standard airline filter capable of removing solid particles and water. This will substantially improve the service life of a following high efficiency filter for removing oil mist. Such a basic filtering system – normally quite adequate for all industrial applications – does not remove odours or gases, hence the importance of the compressor inducting only clean air.

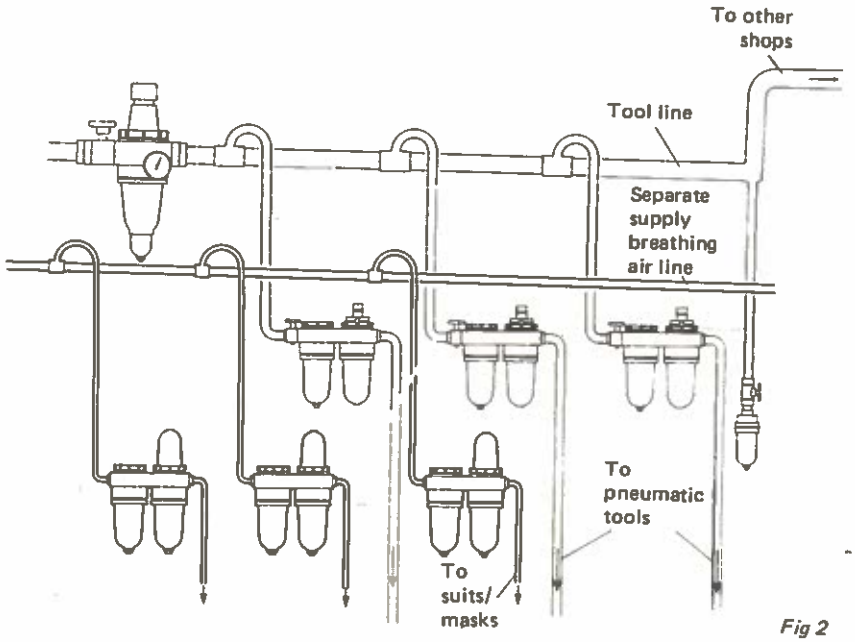


Fig 2

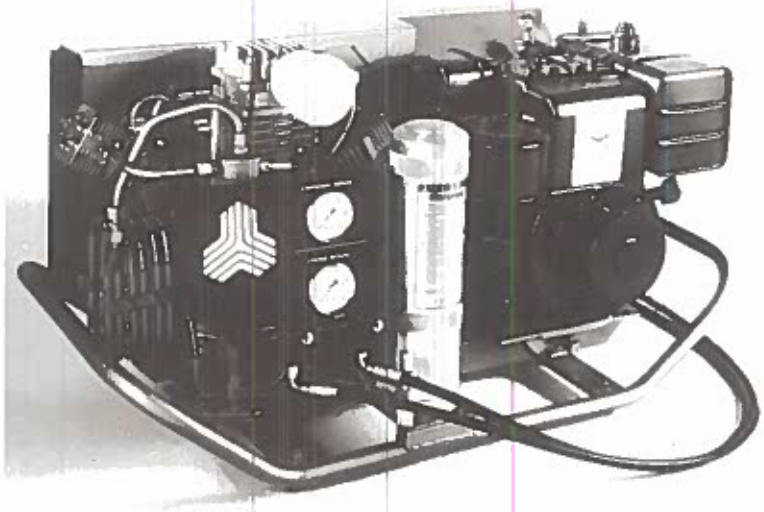
Compressed air installation where personnel are engaged in work requiring a breathing air supply and pneumatic tool supply

Ultra high efficiency filters are required for oil mist removal – *ie* those with an efficiency in excess of 99.9%. This will ensure minimal contamination and also eliminate build-up of oil contamination in downstream pipelines (which can be considerable over a period with, say, a filter having only 99% efficiency).

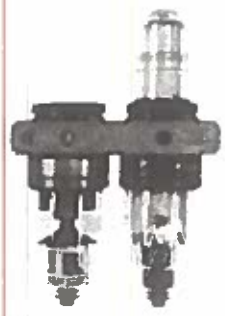
Minimum delivery requirement for breathing air is 4.2 ft^3/m (120 lit/min) free air per person, with a higher figure being desirable. Pipelines, filters and pressure reducing valves must be sized accordingly. Relative humidity of the air should be between 25% and 80% at atmospheric pressure. Acceptable temperature for breathing air is between 15° and 25°C. Humidity requirements largely rule out the use of dryers in the system; alternatively a humidifier may be included in the system if a dryer is essential for other purposes. Continual breathing of dry air can cause discomfort or even respiratory troubles. Certain types of airline lubricators can be used as humidifiers, if necessary (*ie* 'fog' type lubricators which would then generate only water vapour). Temperature requirements may or may not call for cooling equipment.

Odours and Toxic Vapours

If strictly necessary, most odours can be removed by activated charcoal filters, and carbon monoxide by special absorbent media filters. In both cases such filter elements have a limited effective life and need replacement at regular intervals (depending on the concentration of vapours present which they have to absorb). The need for this will become obvious with odour-removing filters, but not so with carbon monoxide which is odourless.



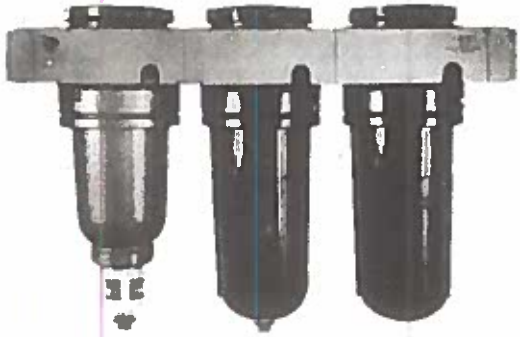
BroomWade breathing air set with petrol engine.



Model M4R-201-A2XD
for one mask.

Model M4R-301-A2XD
for two masks

Norgren breathing air sets.



Model M8R-661-F2XD for four or five masks .

The likelihood of toxic gases being generated within the system is negligible. When found in delivered air, the reason is that the compressor is sited in an area where they can be inhaled. The possible exception is the PTFE ring compressor which, if overheated, could generate toxic vapours. For this reason many authorities specify PTFE ring compressors as unsuitable for providing air for breathing.

Specifically, where toxic gases are suspected or known to be present in the supply from a compressor, the use of respirators or bottled air supplies should be adopted for breathing.

SECTION 6

Data

Standards and Publications

Standard	Subject	Cross Reference(s)
BS21:1973	Pipe threads for tubes and fittings where pressure-tight joints are made on the threads.	ISO 7
BS143 & 1256: 1968	Malleable cast iron and cast copper alloy screwed pipe fittings for steam, air, water, gas and oil.	
BS1123:1976	Specification for safety valves, gauges and other safety fittings for air receivers and compressed air installations.	
BS1387:1967	Steel tubes and tubulars suitable for screwing to BS21 pipe threads	ISO 65
BS1571:	Testing of positive displacement compressors and exhausters.	
Part 1:1975	Acceptance tests.	ISO 1217
Part 2:1975	Simplified acceptance tests for air compressors and exhausters.	
BS1701:1970	Air filters for air supply to internal combustion engines and compressors other than for aircraft.	
BS1710:1975	Identification of pipelines.	ISO/R 508
BS1780:	Bourdon tube pressure and vacuum gauges	
Part 1:1960	Imperial units.	
Part 2:1971	Metric units.	
BS1965	Butt-welding pipe fittings for pressure purposes.	
BS1966:1967	Domed ends for tanks and pressure vessels.	
BS2051:	Tube and pipe fittings for engineering purposes.	
BS1973 Part 1	Copper and copper alloy capillary and compression tube fittings for engineering purposes.	
BS2779:1973	Pipe threads where pressure-tight joints are not made on the threads.	ISO 228
BS2971:	Copper and copper alloys. Tubes.	
BS1972	Tubes for general purposes.	ISO/R 196
BS2917:1977	Specification for graphical symbols used on diagrams for fluid power systems and components (ISO 1219)	ISO 1219 CETOP R3
BS3600:1976	Specification for dimensions and masses per unit length of welded and seamless steel pipes and tubes for pressure purposes.	ISO 336 ISO 64 ISO 1179

Standard	Subject	Cross Reference(s)
BS3641	Symbols for machine tools.	
Part 1:1971	General symbols.	ISO/R 369
Part 3:	Additional general symbols.	ISO/R 369
BS3928:1969	Method for sodium flame test for air filters (other than for air supply to i.c. engines and compressors).	
BS423:1975	Viscosity calssification of industrial liquid lubricants.	ISO 3448
BS4862:1973	Mounting dimensions of single rod double acting pneumatic cylinder, 10 bar	CETOP RP 43P, RP 51P, RP 62P, RP 53P.
BS5118:1975	Rubber air hose	ISO 2398
BS5169:1975	Fusion-welded steel air receivers.	
BS5242:	Tubes for fluid power cylinder barrels	CETOP RP 79P
Part 1:1975	Steel tubes with specially finished bores.	
BS5276:	Pressure vessel details (dimensions)	
Part 4:1977	Standardized pressure vessels.	
BS5319:1976	Specification for quick-release vacuum couplings (screwed and clamp type).	ISO 286/1
BS5344:1976	Methods for acceptance tests for rotary and percussive pneumatic tools	ISO 2787
BS5409	Specification for nylon tubing.	
	Fully plastized nylon tubing types 11 and 12 for use primarily in pneumatic installations.	
BS5500:1976	Unfired fusion welded pressure vessels.	
BS5543:1978	Vacuum technology — graphical symbols	ISO 3753
BS5555:1976	SI units and recommendations for the use of their multiples and of certain other units.	ISO 1000
BS5600:	Powder metallurgical material and products.	
Part 3:1979	Section 3.5 — Determination of bubble test pore size.	ISO 4003

PNEUROP Publications

The following publications are printed in English, French and German; publication number 5601 is printed also in Italian, Spanish and Swedish.

5601	Compressor, Classification and Glossary of Technical Terms.
5602	Pneumatic Tools, Classification and Glossary of Technical Terms.
5603	Vacuum Pumps, Classification and Glossary of Technical Terms.
5604	Measurement of Sound from Pneumatic Equipment.
5605	Flanges and Connections for High Vacuum Equipment, Dimensions.
6602	Vacuum Pumps, Rules of Acceptance, Part 1 (Oil sealed rotary pumps and Roots pumps). (Replaces 5606).
5607	Vacuum Pumps, Rules of Acceptance, Part II (Vapour Pumps).
5608	Vacuum Pumps, Rules of Acceptance, Part III (Turbomolecular Pumps).

5615	Vacuum Pumps, Rules of Acceptance, Part IV (Sputter Ion Pumps).
–	PNEUROP Pamphlets – "Safety in Operation".
–	PNEUROP SC 8 Working Group Report on PNEUROP Sound Test Code (Background and Experience of its application).
–	Graphical Symbols for Control and Instrument Panels on Rock Drilling Rigs.
–	Vibrations in Pneumatic Hand Held Tools – Investigation on Hand-held Grinding Machines – Investigation on Hand-held Percussive Tools
6603/1980	Compressed Air Dryers – Specification and Testing.
660420/1981	CAGI/PNEUROP – Recommendation for Performance Statements for Packaged Air Compressors of Displacement Type.
6606/1981	Vacuum Flanges and Connexions. Dimensions.
6607/1980	A simplified method for air volume flow rate by means of circular flow conditions.
–	Test procedure for the measurement of vibration from hand held (portable) power driven grinding machines.
–	A comparison of Sound Power Levels from Portable Air Compressors Based Upon Test Methodologies Adopted by U.S. EPA and the CEC.

The following publications are printed only in English; German editions are in preparation.

–	PNEUROP Committee No.8. Noise Levels of Pneumatic Equipment used on building sites. Regulations in Europe and USA relating to such noise. (Issued by BCAS in 1975).
5611-2	Code of Practice, Reciprocating Compressors for the Process Industry.
5612-2	Data Sheets for 5611-2.
5613-2	Code of Practice, Turbo Compressors for the Process Industry.
5614-2	Data Sheets for 5613-2.
5618-2	Code of Practice, Screw and Related Compressors for the Process Industry.
5619-2	Data Sheets for 5618-2.

CETOP Publications

The following documents are printed in English, French and German; those marked "(PN)" are joint PNEUROP/CETOP recommendations; documents without the suffix P apply equally to hydraulics.

R.2	Classification of Fluid Power Terms and Documents.
RP.4P	Pneumatic Cylinders, Suggested data for inclusion as a minimum in Manufacturers' Technical Sales Literature.
RP.5P	Specification for Pneumatic Cylinders.
RP.6P	Pipe Couplings for Pneumatic Piping – Coupling Thread.
RP.7P	Pneumatic Cylinders, Recommended minimum relation of port size (thread) to cylinder bore. (In part superseded by ISO R.1939).
RP.19 to RP.30P	Recommended data for inclusion in manufacturers' technical sales literature.
RP.19P	Pneumatic directional control valves.
RP.20P	Pneumatic flow control valves.
RP.21P	Pneumatic pressure control valves.
RP.22P	Pneumatic shuttle, non-return and quick exhaust valves.
RP.23P	Pneumatic pressure intensifiers.

RP.24P	Pneumatic Rectilinear piston type cylinders.
RP.25P	Pneumatic filters and water traps.
RP.26P	Lubricators.
RP.27P	Air Dryers.
RP.28P	Connections.
RP.29P	Pneumatic Quick-action couplings.
RP.30P	Pneumatic Rotating and Telescopic joints.
RP.32P	Subplates for Pneumatic Directional Control Valves.
RP.33	Graphical Symbols and definitions for operations of logic and related functions in fluid logic circuits.
RP.34P	Couplings for Industrial Air Hoses — 10 bar.
RP.37	Recommended diameters for pneumatic tubes and hoses. (Under revision).
RP38P	Guidance on relation between port threads and pipe hose diameters. (Under revision).
RP.40P and amendment	Hose Couplings, claw type.
RP.41	Hydraulic and Pneumatic Circuits, circuit diagram.
RP.43P	Pneumatic Cylinders 10 bar, Mounting dimensions (bores 32–100 mm).
RP.49P	Technological Symbols for Fluid Logic and related devices with and without moving parts.
RP.50P	Flow capacity value of Pneumatic components. (Under revision).
RP.51P	Pneumatic cylinders; Basic data.
RP.52P	Pneumatic cylinders, Operating conditions and dimensions (bores 8 to 25 mm).
RP.53P	Pneumatic cylinders, Operating conditions and dimensions (bores 125 to 320 mm).
RP.54P	Specification for Polyamide Tubing 11 and 12 bar for Pneumatic Transmissions.
RP.57P	Pressure Relief Valves — Recommended data for inclusion in manufacturers' technical sales literature.
RP.59P	Quick Action Couplings — plug dimensions —10 bar. Supplements Nos. 1 & 2 (1975).
RP.68	Identification Code of Ports and Operators of Pneumatic and Hydraulic Control Valves and Other Components. (Under review).
RP.71	Quantities, Symbols and Units of the International System (SI) to be used for Fluid Power.
RP.76	Outside Diameters for Tubes in Fluid Power Applications.
RP79P	Specification for non-ferrous pneumatic cylinder tube (ready to use).
RP.80	Cone Type Connection — 24° — For Fluid Power Tubes and Hoses.
RP.81H	Compatibility of Hydraulic Fluids with Elastomeric Materials.
RP.82P	Response Time Characteristics of Pneumatic Directional Control Valves.
RP.83P	Characteristics of the Pressure Medium to be Supplied to Pneumatic, Fluid Logic and Fluidic Devices and Systems.
RP.84P	Flow Co-efficients of Pneumatic Components.
RP.85P	Characteristics of Pneumatic Components
RP.89P	Clevis for Pneumatic Cylinders.
RP.90P	Rod end attachment (spherical) eye.
RP.100 1976	Hydraulic and Pneumatic Fluid Power Glossary
RP.101	Lexicon of Terms from Hydraulic and Pneumatic Fluid Power Glossary RP.100P.
RP.102P	Rod end clevis for pneumatic cylinders.
RP.103P	Rod end bearing for pneumatic cylinders.

Publications Pending (1982)

PNEUROP	Compressed Air Dryers, Specification and Testing (3 languages).
	Vacuum Pumps, Classification and Glossary of Technical Terms (7 languages).
	A Simplified Method for Air Volume Flow Rate Measurement by means of Circular Arc Venturi Nozzles at Critical Flow Conditions.
	Proposed ANNEXE TO ISO 1217, Displacement Compressor Acceptance Test.
PNEUROP/ CAGI	Recommendation for Performance Statements for Packaged Air Compressors of Displacement Type.
	Recommendation for Performance Corrections for Liquid Injected Displacement Type Rotary Compressor.

BCAS Publications

Guide to the Selection and Installation of Compressed Air Services.

Buyers' Guide to Compressed Air Plant and Equipment.

BCAS Brochure.

The Principal SI Units to be used by the Compressed Air Industry.

A Guide to Compressor Noise Reduction.

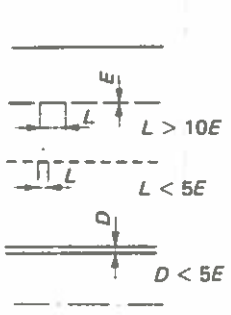
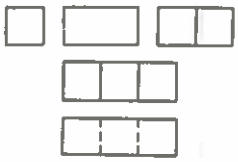
GRAPHICAL SYMBOLS FOR PNEUMATIC SYSTEMS AND COMPONENTS

THE FOLLOWING symbols are specified in ISO 1219:1976 (E/F). They are virtually identical to those specified in BS2917. The other standard source of reference for Europe, CETOP RP3, is superceded by ISO 1219.

General (Basic and Functional Symbols)

The symbols for hydraulic and pneumatic equipment and accessories are *functional* and consist of one or more *basic symbols* and in general of one or more *functional symbols.* The symbols are neither to scale nor in general orientated in any particular direction.

BASIC SYMBOLS

Description	Application	Symbol
Line: – continuous – long dashes – short dashes	flow lines	
– double	Mechanical connections (shafts, levers, piston-rods)	
– long chain thin (optional use)	Enclosure for several components assembled in one unit	
		L = length of dash *E* = thickness of line *D* = space between lines
Circle, semi-circle	As a rule, energy conversion units (pump, compressor, motor)	
	Measuring instruments	
	Non-return valve, rotary connection, etc	
	Mechanical link, roller, etc	
	Semi-rotary actuator	
Square, rectangle	As a rule, control valve(s) except for non-return valves	
Diamond	Conditioning apparatus (filter, separator, lubricator, heat exchanger)	

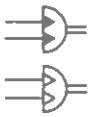
BASIC SYMBOLS

Description	Application	Symbol
	Flow line connection	$d \approx 5E$
		E = thickness of line
	Spring	
	Restriction: – affected by viscosity	
	– unaffected by viscosity	

FUNCTIONAL SYMBOLS

Description	Application	Symbol
Triangle	The direction of flow and the nature of the fluid	
– solid	Hydraulic flow	
– in outline only	Pneumatic flow or exhaust to atmosphere	
Arrow	Indication of: – direction	
	– direction of rotation	
	– path and direction of flow through valves	
	For regulating apparatus as in *Pressure Control Valves* both representations with or without a tail to the end of the arrow are used without distinction. As a general rule the line perpendicular to the head of the arrow indicates that when the arrow moves the interior path always remains connected to the corresponding exterior path.	
Sloping arrow	Indication of the possibility of a regulation or of a progressive variability	

PUMPS AND COMPRESSORS

Description	Remarks	Symbol
Fixed capacity compressor		
Fixed capacity pneumatic motor: – with one direction of flow – with two directions of flow		
Variable capacity pneumatic motor: – with one direction of flow – with two directions of flow		
Oscillating motor: – hydraulic – pneumatic		
Pump/motor units	Unit with two functions, either as pump or as rotary motor	
Fixed capacity pump/motor unit: – with reversal of the direction of flow – with one single direction of flow – with two directions of flow	Functioning as pump or motor according to direction of flow Functioning as pump or motor without change of direction of flow Functioning as pump or motor with either direction of flow	
Variable capacity pump/motor unit: – with reversal of the direction of flow – with one single direction of flow – with two directions of flow		

CYLINDERS

Description	Remarks	Symbol	
		Detailed	Simplified
Single-acting cylinder:	Cylinder in which the fluid pressure always acts in one and the same direction (on the forward stroke)		
– returned by an unspecified force	General symbol when the method of return is not specified		
– returned by spring			
Double acting cylinder	Cylinder in which the fluid pressure operates alternately in both directions (forward and backward strokes)		
– with single piston rod			
– with double-ended piston rod			
Differential cylinder	The action is dependent on the difference between the effective areas on each side of the piston		
Cylinder with cushion:			
– with single fixed cushion	Cylinder incorporating fixed cushion acting in one direction only		
– with double fixed cushion	Cylinder with fixed cushion acting in both directions		
– with single adjustable cushion			
– with double adjustable cushion			
Telescopic cylinder:			
– single acting	The fluid pressure always acts in one and the same direction (on the forward stroke)		
– double acting	The fluid pressure operates alternately in both directions (forward and backward strokes)		

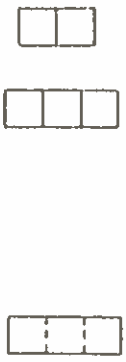
PRESSURE INTENSIFIERS

Description	Remarks	Symbol	
		Detailed	Simplified
– for one type of fluid	eg a pneumatic pressure x is transformed into a higher pneumatic pressure y		
– for two types of fluid	eg a pneumatic pressure x is transformed into a higher hydraulic pressure y		
Air-oil actuator	Equipment transforming a pneumatic pressure into a substantially equal hydraulic pressure or vice versa		

CONTROL VALVES

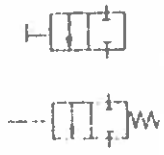
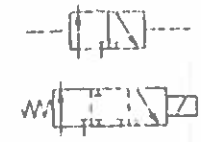
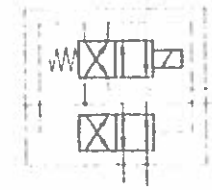
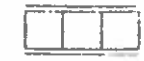
Description	Remarks	Symbol
Method of representation of valves	Made up of one or more squares and arrows *In circuit diagrams hydraulic and pneumatic units are normally shown in the unoperated condition*	
One single square	Indicates unit for controlling flow or pressure, having in operation an infinite number of possible positions between its end positions so as to vary the conditions of flow across one or more or its ports, thus ensuring the chosen pressure and/or flow with regard to the operating conditions of the circuit	
Two or more squares	Indicate a directional control valve having as many distinct positions as there are squares. The pipe connections are normally represented as connected to the box representing the unoperated condition. The operating positions are deduced by imagining the boxes to be displaced so that the pipe connections correspond with the ports of the box in question	
Simplified symbol for valves in cases of multiple repetition	The number refers to a note on the diagram in which the symbol for the valve is given in full	

DIRECTIONAL CONTROL VALVES

Description	Remarks	Symbol
Flow paths: – one flow path – two closed ports – two flow paths – two flow paths and one closed port – two flow paths with cross connection – one flow path in a by-pass position, two closed ports	Square containing interior lines	
Non-throttling directional control valve	The unit provides distinct circuit conditions each depicted by a square Basic symbol for 2-position directional control valve Basic symbol for 3-position directional control valve A transitory but significant condition between two distinct positions is optionally represented by a square with dashed ends A basic symbol for a directional control valve with two distinct positions and one transitory intermediate condition	
Designation: The first figure in the *designation* shows the number of ports (excluding pilot ports) and the second figure the number of distinct positions		

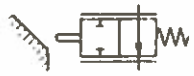
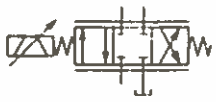
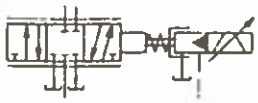
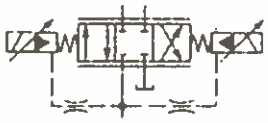
cont.....

DIRECTIONAL CONTROL VALVES (contd.)

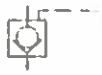
Description	Remarks	Symbol
Directional control valve 2/2: – with manual control – controlled by pressure operating against a return spring (eg on air unloading valve)	Directional control valve with 2 ports and 2 distinct positions	
Directional control valve 3/2: – controlled by pressure in both directions – controlled by solenoid with return spring	Directional control valve with 3 ports and 2 distinct positions	
Directional control valve 4/2: – controlled by pressure in both directions by means of a pilot valve (with a single solenoid and spring return)	Directional control valve with 4 ports and 2 distinct positions	Detailed  Simplified
Directional control valve 5/2: – controlled by pressure in both directions	Directional control valve with 5 ports and 2 distinct positions	
Throttling directional control	The unit has 2 extreme positions and an infinite number of intermediate conditions with varying degrees of throttling All the symbols have parallel lines along the length of the boxes. Showing the extreme positions Showing the extreme positions and a central (neutral) position	 

cont.....

DIRECTIONAL CONTROL VALVES (contd.)

Description	Remarks	Symbol
Throttling directional control (contd.)		
– with 2 ports (one throttling orifice)	For example: Tracer valve plunger operated against a return spring	
– with 3 ports (two throttling orifices)	For example: Directional control valve controlled by pressure against a return spring	
– with 4 ports (four throttling orifices)	For example: Tracer valve, plunger operated against a return spring	
Electro-hydraulic servo valve: Electro-pneumatic servo valve:	A unit which accepts an analogue electrical signal and provides a similar analogue fluid power output	
– single stage	– with direct operation	
– two stage with mechanical feedback	– with indirect pilot operation	
– two stage with hydraulic feedback	– with indirect pilot operation	

NON-RETURN VALVES, SHUTTLE VALVE, RAPID EXHAUST VALVE

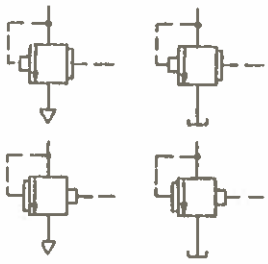
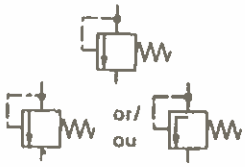
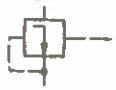
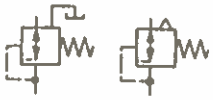
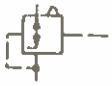
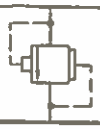
Description	Remarks	Symbol
Non-return valve:		
— free	Opens if the inlet pressure is higher than the outlet pressure	
— spring loaded	Opens if the inlet pressure is greater than the outlet pressure plus the spring pressure	
— pilot controlled	With pilot control it is possible to prevent: — closing of the valve	
	— opening of the valve	
— with restriction	Unit allowing free flow in one direction but restricted flow in the other	
Shuttle valve	The inlet port connected to the higher pressure is automatically connected to the outlet port while the other inlet port is closed	
Rapid exhaust valve	When the inlet port is unloaded the outlet port is freely exhausted	

PRESSURE CONTROL VALVES

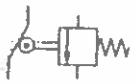
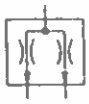
Description	Remarks	Symbol
Pressure control valve: — 1 throttling orifice normally closed		
— 1 throttling orifice normally open		
— 2 throttling orifices, normally closed		
Pressure relief valve (safety valve):	Inlet pressure is controlled by opening the exhaust port to the reservoir or to atmosphere against an opposing force (for example a spring)	
— with remote pilot control	The pressure at the inlet port is limited, or to that corresponding to the setting of a pilot control	

cont.....

PRESSURE CONTROL VALVES (contd.)

Description	Remarks	Symbol
Proportional pressure relief	Inlet pressure is limited to a value proportional to the pilot pressure	
Sequence valve	When the inlet pressure overcomes the opposing force of the spring, the valve opens permitting flow from the outlet port	
Pressure regulator or reducing valve (reducer of pressure):	A unit which, with a variable inlet pressure, gives substantially constant output pressure provided that the inlet pressure remains higher than the required outlet pressure	
– without relief port		
– without relief port with remote control	Outlet pressure is dependent on the control pressure	
– with relief port		
– with relief port, with remote control	Outlet pressure is dependent on the control pressure	
Differential pressure regulator	The outlet pressure is reduced by a fixed amount with respect to the inlet pressure	
Proportional pressure regulator	The outlet pressure is reduced by a fixed ratio with respect to the inlet pressure	

FLOW CONTROL VALVES

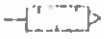
Description	Remarks	Symbol	
Throttle valve:	Simplified symbol (Does not indicate the control method or the state of the valve)		
– with manual control	Detailed symbol (indicates the control method or the state of the valve)		
– with mechanical control against a return spring (braking valve)			
Flow control valve:	Variations in inlet pressure do not affect the rate of flow	Detailed	Simplified
– with fixed output			
– with fixed output and relief port to reservoir	With relief for excess flow		
– with variable output			
– with variable output and relief port to reservoir	With relief for excess flow		
Flow dividing valve	The flow is divided into two flows in a fixed ratio substantially independent of pressure variations		
Shut-off valve	Simplified symbol		

SOURCES OF ENERGY

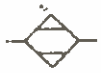
Description	Remarks	Symbol
Pressure source	Simplified general symbol	
Hydraulic pressure source Pneumatic pressure source	Symbols to be used when the nature of the source should be indicated	
Electric motor	Symbol 113 in IEC Publication 117.2	
Heat engine		
Flow lines and connections Flow line: – working line, return line and feed line – pilot control line – drain or bleed line – flexible pipe – electric line	Flexible hose, usually connecting moving parts	
Pipeline junction		
Crossed pipelines	Not connected	
Air bleed		
Exhaust port: – plain with no provision for connection – threaded for connection		

cont.....

SOURCES OF ENERGY (contd.)

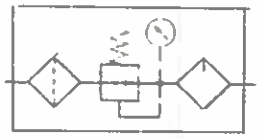
Description	Remarks	Symbol
Power take-off:	On equipment or lines, for energy take-off or measurement	
– plugged		
– with take-off line		
Quick-acting coupling:		
– connected, without mechanically opened non-return valve		
– connected, with mechanically opened non-return valves		
– uncoupled, with open end		
– uncoupled, closed by free non-return valve		
Rotary connection:	Line junction allowing angular movement in service	
– one way		
– three way		
Silencer		

RESERVOIRS

Description	Remarks	Symbol
Reservoir open to atmosphere:		
— with inlet pipe above fluid level		
— with inlet pipe below fluid level		
— with a header line		
Pressurized reservoir		
Accumulators	The fluid is maintained under pressure by a spring, weight or compressed gas (air, nitrogen, etc)	
Filter or strainer		
Water trap: — with manual control		
— automatically drained		
Filter with water trap: — with manual control		
— automatically drained		
Air dryer	A unit drying air (for example by chemical means)	

cont.....

RESERVOIRS (contd.)

Description	Remarks	Symbol
Lubricator	Small quantities of oil are added to the air passing through the unit, in order to lubricate equipment receiving the air	
Conditioning unit	Consisting of filter, pressure regulator, pressure gauge and lubricator – Detailed symbol – Simplified symbol	
Heat exchangers	Apparatus for heating or cooling the circulating fluid	
Temperature controller	The fluid temperature is maintained between two pre-determined values. The arrows indicate that heat may be either introduced or dissipated	
Cooler	The arrows in the diamond indicate the extraction of heat – without representation of the flow lines of the coolant – indicating the flow lines of the coolant	
Heater	The arrows in the diamond indicate the introduction of heat	

CONTROL MECHANISMS

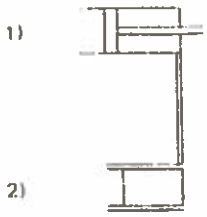
Description	Remarks	Symbol
Rotating shaft: — in one direction — in either direction	The arrow indicates rotation	
Detent	A device for maintaining a given position	
Locking device	*The symbol for unlocking control is inserted in the square	
Over-centre device	Prevents the mechanism stopping in a dead centre position	
Pivoting devices: — simple — with traversing lever — with fixed fulcrum		

CONTROL METHODS

Description	Remarks	Symbol
Muscular control:	General symbol (without indication of control type)	
– by push button		
– by lever		
– by pedal		
Mechanical control:		
– by plunger or tracer		
– by spring		
– by roller		
– by roller, operating in one direction only		
Electrical control:		
– by solenoid	– with 1 winding	
	– with 2 windings operating in opposite directions	
	– with 2 windings operating in a variable way progressively, operating in opposite directions	
– by electric motor		
Control by application or release of pressure		
Direct acting control:		
– by application of pressure		

cont.....

CONTROL METHODS (contd.)

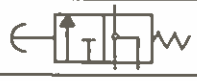
Description	Remarks	Symbol
– by release of pressure		
– by different control areas	In the symbol the larger rectangle represents the larger control area, ie the priority phase	
Indirect control, pilot actuated:	General symbol for pilot directional control valve	
– by application of pressure		
– by release of pressure		
Interior control paths	The control paths are inside the unit	
Combined control:		
– by solenoid and pilot directional valve	The pilot directional valve is actuated by the solenoid	
– by solenoid or pilot directional valve	Either may actuate the control independently	
Mechanical feedback	The mechanical connection of a control apparatus moving part to a controlled apparatus moving part is represented by the symbol which joins the two parts connected	 1) Controlled apparatus 2) Control apparatus

SUPPLEMENTARY EQUIPMENT

Description	Remarks	Symbol
Pressure measurement: – pressure gauge	The point on the circle at which the connection joins the symbol is immaterial	
Temperature measurement: –Thermometer	The point on the circle at which the connection joins the symbol is immaterial	
Measurement of flow: – Flow meter – Integrating flow meter		
Other apparatus Pressure electric switch		

VDMA SYMBOLS

VDMA symbols for valve actuation are shown below. This graphic system was originated by the Organisation of German Machine Manufacturing Industry specifically for hydraulic valves, but has been widely used in Europe for pneumatic as well as hydraulic components.

Actuation Method	Symbol	Examples	
Hand lever			Direct actuated hand valve
Push Button			Direct actuated push-button (spring return)
Treadle			Direct actuated foot valve (spring return)
Cam Follower			Cam valve
Air or Fluid Actuator			3-position, air actuated valve
Solenoid			Solenoid (direct) actuated valve
Other Method			The asterisk in the first box refers to a footnote. For example, this footnote would say "photo electric cell"
Indirect actuation (basic symbol)			Solenoid actuated, pilot operated valve

Auxiliary Terms

Spring			Foot actuated valve with spring return
Adjustment			Adjustable spring load
Detent			Direct operated hand valve, spring returned with detent
Ratchet (Pawl)			The asterisk means a footnote will explain the ratchet.

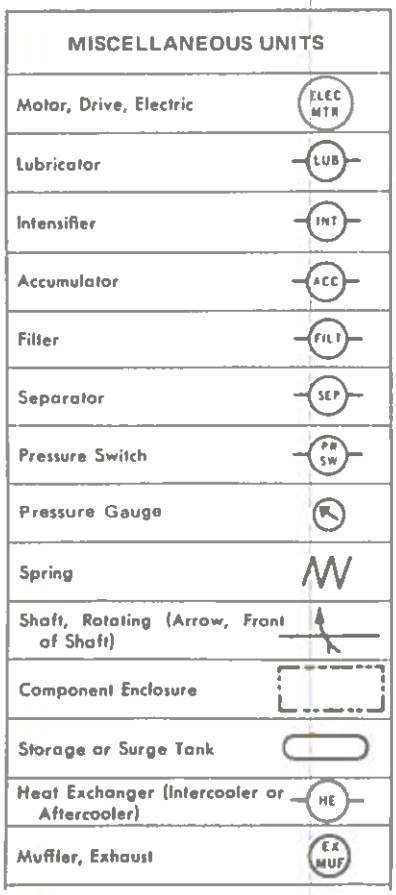
JIC GRAPHICAL SYMBOLS

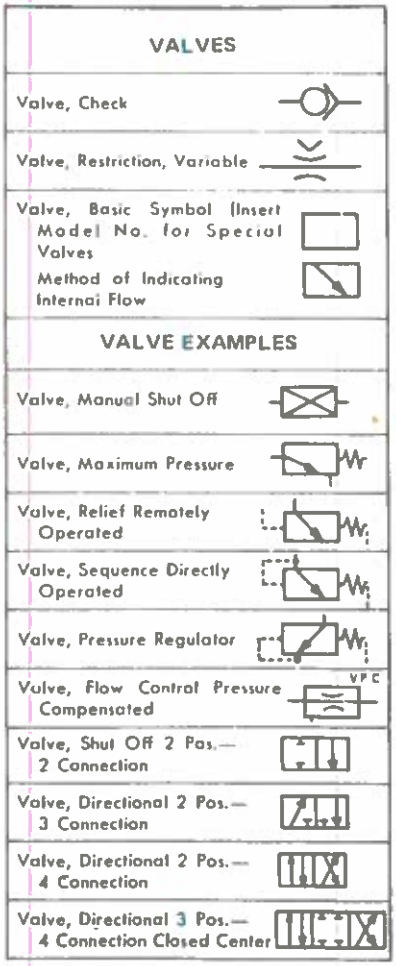
The Joint Industry Conference (JIC) graphical pneumatic symbols are widely used by the American pneumatic industry (and also elsewhere). The system is based on graphic symbols emphasizing both the function of a component and its construction.

The basic symbols are divided into three main categories — supply, control and work element. The basic valve symbol consists of one or more square envelopes, with lines to show flow direction between ports. Direction of flow paths is indicated by arrows. The valve is drawn to show normal or neutral position. A single envelope symbol is used for valves with only one flow path. Multiple envelopes are used where there is more than one flow path. Envelopes move to show how flow paths change when the valve elements are shifted to various positions.

Actuation method is indicated in a small rectangular box attached to the end of the envelope opposite the flow side. When there are two actuators, one box is shown at each end.

External air lines are designated by solid lines. Crossover of air lines is represented by a half circle. Pilot lines are shown by dashed lines.




VALVES
Valve, Check
Valve, Restriction, Variable
Valve, Basic Symbol (Insert Model No. for Special Valves
Method of Indicating Internal Flow
VALVE EXAMPLES
Valve, Manual Shut Off
Valve, Maximum Pressure
Valve, Relief Remotely Operated
Valve, Sequence Directly Operated
Valve, Pressure Regulator
Valve, Flow Control Pressure Compensated
VPC
Valve, Shut Off 2 Pos.— 2 Connection
Valve, Directional 2 Pos.— 3 Connection
Valve, Directional 2 Pos.— 4 Connection
Valve, Directional 3 Pos.— 4 Connection Closed Center

LINES

- Line, Working
- Line, Pilot
- Exhaust
- Line, Flexible
- Connector (Dot to be 3 x width of Associated Line)
- Direction of Flow
- Line, Passing
- Line, Joining (Tee Cross, etc.) (Dot to be 3 x w)
- Plug or Plugged Connection
- Testing Station (Gauge Connection) — T, S
- Power Take-Off — P, T, O
- Restriction, Fixed

MOTORS AND CYLINDERS

- Motor, Rotary Fixed Displacement — MF
- Motor, Rotary Variable Displacement — MV
- Motor, Oscillating — MO
- Cylinder, Single Acting
  - Plunger Type
  - Piston Type
- Cylinder, Double Acting
  - Single End Rod
  - Double End Rod
- Cylinder, Rotating, Air — R

COMPRESSORS

- Compressor, Single Fixed Displacement — CF
- Compressor, Single Variable Displacement — CV

METHODS OF OPERATION

- Control, Basic Symbol
- Control, Centrifugal — CENT
- Control, Compensator — COMP
- Control, Compensator Pressure — PR COMP
- Control, Compensator Temperature — TEMP CONT
- Control, Cylinder — CYL
- Control, Detent — DET
- Control, Manual — MAN
- Control, Mechanical — MECH
- Control, Motor Electric — ELEC MTR
- Control, Motor Air — AIR MTR
- Control, Pilot Hydraulic — HYD
- Control, Pilot Air — AIR
- Control, Servo — SERVO
- Control, Solenoid — SOL
- Control, Solenoid Hyd. Pilot Operated — SOL HYD
- Control, Solenoid Air Pilot Operated — SOL AIR
- Control, Thermal — THERM
- Control, Pilot Hyd. Differential Area — HYD DIFF

CETOP RECOMMENDATIONS FOR CIRCUIT DIAGRAMS

The following rules and recommendations are extracted from CETOP Provisional Recommendations RP41 — Hydraulic and Pneumatic Systems Circuit Diagrams — published by the Comite Europeen des Transmissions Oleohydrauliques et Pneumatiques.

The purpose of RP41 is to facilitate the design, construction, description and maintenance of fluid power systems by aiding communication through uniform presentation and by preventing confusion. The recommendation is based on the applicable ISO standards and CETOP recommendations. It is not necessary for a circuit diagram to incorporate all the requirements listed but anything which is shown must be in accordance with the appropriate sections of the document. The recommendation is generally applicable to the representation of all fluid power systems.

Design of Circuit Diagrams

The circuit diagram must show fluid circulation for all the control and motion functions. All hydraulic and pneumatic components and flow line junctions in the plant will be represented functionally. In addition all control components that affect the operational processes of the fluid power system will be shown.

The circuit diagram need not be constructed with reference to the physical arrangement of the equipment in the system. Interdependent circuits should be shown on one diagram. For electro-hydraulic or electro-pneumatic controls the circuit diagram will, however, be divided into one hydraulic or pneumatic circuit diagram and, if necessary, a separate electrical circuit.

The following circuit diagram formats should be used:—

ISO-A-Format (ISO Recommendation R216)
Length: preferably 297 mm
exceptionally 420 or 594 mm
Width: up to 1189 mm
Diagrams will be folded to ISO A4 format with binding margin (except for diagrams for reproduction).

The layout of the circuit diagram must be clear. Care should be taken to see that the circuit paths are easily followed. Pneumatic circuits should be drawn if possible in the sequence of the operational process of function. The components of the individual control chains and groups should be drawn wherever possible in the direction of energy flow.

Cylinders and directional control valves should preferably be shown horizontally. Fluid lines should be represented wherever possible by straight lines and without intersections.

The location of control components such as sequence switch cams and simple cams, etc should be indicated. If a signalling element is to be actuated by a one way trip, then the actuating direction will be indicated by an arrow. In the case of electro-hydraulic and electro-pneumatic control systems, signalling and final control elements such as limit switches and electromagnetic valves will be shown in both circuit diagrams.

Individual sub-circuits may be identified. The function of each operating element should be indicated beside it (eg clamping, lifting, etc). Where necessary for better understanding a function chart can be drawn up.

Components should be represented by symbols in accordance with the applicable standards, eg ISO R1219 for fluid power equipment. Where no symbols exist, the operation of the part should be clearly shown, eg by means of a simplified sectional representation or the like.

Symbols that are repeated in the diagram can be represented by a numbered rectangle and the corresponding symbols shown separately.

Data to go on Circuit Diagram

Every component must be clearly identified, eg by item number. Components in a given sub-circuit may be given a common prime identification plus a running sub-number, eg sub-circuit 2 component 3 = 2.3.

Component Operating Positions

In the case of valves with discrete switching positions these positions may be identified by letters, the letter 'O' being used only for the spring returned position (eg springloaded valves).

Identification of Ports

Ports will be identified on the circuit diagram by the characters marked on the components or the connection plate. This applies to individual components and sub-assemblies.

Identification of Oil or Air Lines

The pipelines are to be drawn according to their function in line with ISO Recommendations ISO R1219 (5.2.1).

In cases where hydraulic and pneumatic circuits are represented on one diagram they should be differentiated in accordance with ISO R1219.

Should it be necessary to identify the pipelines more precisely than in ISO R1219, the following code applies:—

full line	———————	pressure line
dashed line	– – – – – – – – – –	pilot control line
short dotted line	- - - - - - - - - - - - - - - - - -	drain or bleed line
full line with short cross lines	⊣⊣⊣⊣⊣⊣⊣⊣⊣⊣⊣⊣⊣⊣⊣⊣⊣	return or exhaust line
full line with crosses	✕✕✕✕✕✕✕✕✕✕✕✕✕	replenishing line
full line with arrows	➢➢➢➢➢➢➢➢➢➢➢➢	pump inlet line

If pipelines are identified by colour, the following colour code should be used:—

red line	pressure line
red dashed line	pilot control line
blue line	return or exhaust line
blue dashed line	drain or bleed line
green line	replenishing line
yellow line	pump inlet line

Any deviations from the above or additional colours should be explained on the circuit diagram.

The following technical data should be given beside each individual component or on a component list.

For reservoirs, give the volume represented by the maximum quantity of fluid in the system and, if required, the minimum quantity, the latter figure indicating the point at which replenishment is necessary. Show also the type and viscosity grade of the fluid. For pneumatic reservoirs indicate the volume.

For constant delivery pumps, indicate nominal delivery rate for the application and, for variable delivery pumps, minimum and maximum delivery rate. In both cases show speed of rotation and nominal power of the prime movers. For prime movers indicate nominal power at stated speed and speed range if applicable. For pressure valves and pressure switches, show pressure setting or permissible pressure range for the system.

For cylinders, the internal diameter of cylinder, the diameter of piston rod, maximum stroke in mm, eg 100/50 x 500, and if necessary speed and force should be given. For each cylinder, describe the function, eg clamping, lifting, transverse feed etc. For telescopic cylinders this information should be given for each stage. Where necessary the circuit should be supplemented by a stroke-time diagram showing the speed.

For semi rotary actuators show torque in relation to pressure, if necessary in each direction, the angular movement, the speed and the function.

For fixed displacement motors, the nominal capacity and, for variable displacement motors, the maximum capacity per revolution should be indicated. For both give the torque in relation to pressure for the application, speed range, direction of rotation and function.

For gas loaded hydraulic accumulators indicate the precharge pressure, type of gas, effective gas volume, working pressure range and corresponding draw off capacity. See RP62 H when available.

For pipes, give the external diameter in millimetres, wall thickness in millimetres and pipe material if necessary. For flexible hose, indicate the nominal diameter and type.

For delaying devices the time delay in seconds or range of adjustment should be shown. Any temporary connection ports such as bleed and test points should be identified by size, type and as marked on component.

For filters give the nominal filtration rating, flow rate and nominal pressure.

STANDARD ELECTRICAL SYMBOLS
(Based on IEC Publication 117.2)

conduit, multi-core line

direct current (D.C.)

alternating current (A.C.)

earth connection

limiting line

branching

crossing point without electric connection

fuse, the side under tension is marked

socket-contact

contact pin

pilot lamp

diode

three-phase delta connection (D-connection)

three-phase star connection (Y-connection)

electric motor

three-phase transformer

switch

two-way switch with interruption

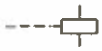
push button switch

switch, manually-operated

switch, electromagnetically-operated

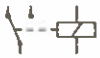
switch, thermally-operated

pressure switch with contact closing when the pressure exceeds a pre-set value


electromagnetic operation

thermal operation

one-coil relay

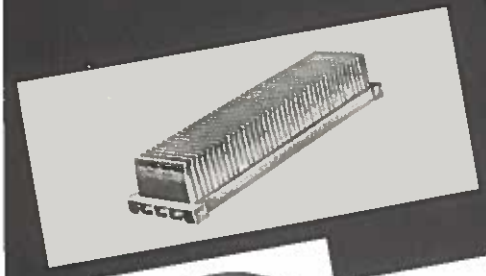

one-coil relay with two-way switch with interruption

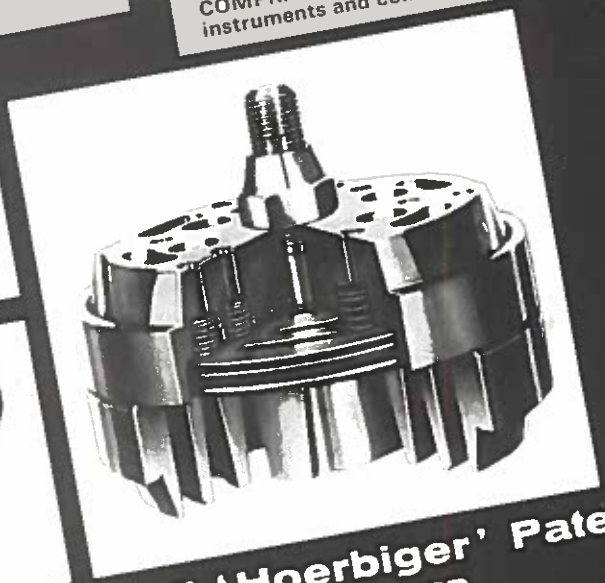
SECTION 7

COMPRAX Automatic Valves

Automatic Plate Valves for air and gas compressors, pumps and blowers. Special high-lift types for non-return valves in pipelines. A comprehensive range of valves is available to suit all operating conditions. Frictionless guided valves for non-lubricated machines a speciality. Extra strong high pressure valves and enclosed safety guards. Valves to customers' designs to suit particular requirements. Spare parts for all valve types. Streamline valves for 'Sulzer' Diesel Engines. Pilot Valves and Regulating Monitoring Equipment for Compressors.

Now also the complete range of COMPRAX HOERBIGER Pneumatic instruments and controls.



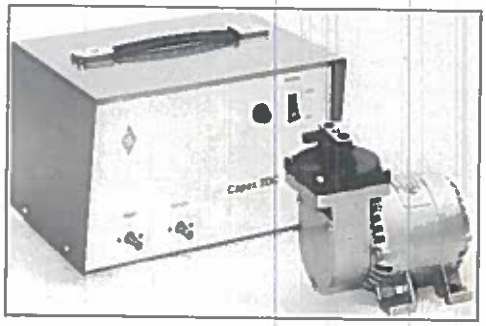


Sole Licensees of all 'Hoerbiger' Patents

COMPRESSOR ACCESSORIES LTD.

1649, Pershore Road, Strichley
Birmingham B30 3DR England
Telephone: 021—458—1124
Telex: 338835 Compax Bham

OIL FREE
DIAPHRAGM PUMPS FOR AIR + GAS PRESSURE OR VACUUM APPLICATIONS



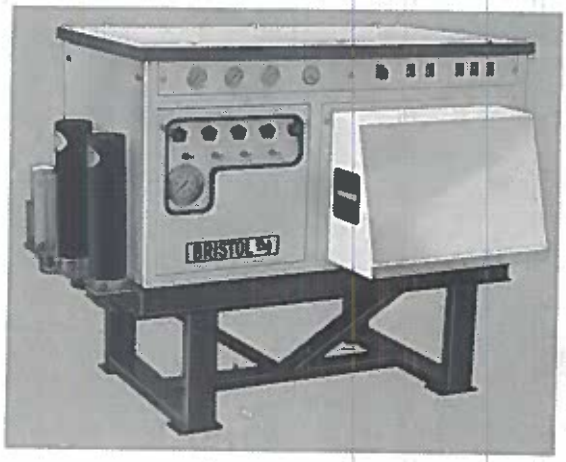
CAPEX 2D & 2DC

- QUIET IN OPERATION
- SMALL PHYSICAL SIZE
- CORROSION RESISTANT MODELS AVAILABLE
- EASY TO SERVICE
- A CHOICE OF 7 DIFFERENT MODELS

Charles Austen Pumps Ltd.

100 ROYSTON ROAD, BYFLEET,
WEYBRIDGE, SURREY KT14 7PB.
TEL: 09323 43224 TELEX: 928368

HIGH PRESSURE AIR YOU CAN RELY ON.



Pressures up to 414 bar and outputs up to 8.6 litre/sec - Oil carryover less than 1ppmW with water down to 0.007mg/litre. Bristol four stage airflow-cooled compressors and purification systems offer you all this with proven reliability in worldwide climatic conditions.

Our range also includes compressors up to 69 bar and 11 litre/sec and workshop sets up to 13 bar and 19 litre/sec.

BRISTOL

BRITISH MADE

Bristol Pneumatic Ltd., Lodge Causeway, Bristol BS16 3JS. Tel: 0272 657511. Telex: 44641.

Survey of Compressors

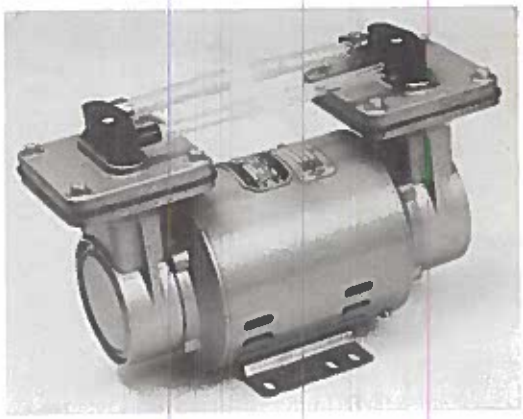
Charles Austen Pumps Ltd

The company's products include a range of 'oil free' diaphragm pumps for flow rates from 0 to 70 lit/min, pressure from 0 to 4 kg/cm^2, or vacuum of 0 to 750 mm Hg. These pumps are used extensively to sample air and gas for process control instruments. Diaphragm liquid pumps capable of 50 to 800 cm^3/min and centrifugal liquid pumps covering flow rates up to 1 600 lit/hr are also available.

OIL FREE DIAPHRAGM PUMPS FOR AIR AND GAS, PRESSURE OR VACUUM

Type	Height mm	Width mm	Length mm	Weight kg	Free Flow lit/min	Vacuum mm Hg	Pressure kg/cm^2	Motor Standard Pump	Motor Consumption in Amps (Std.Pumps)
L11	83	53	68	0.18	1.5	150	0.3	G	0.25 (6 v)
L12	83	53	51	0.2	0.2	200	0.3	E	0.020
DA1 SE	90	62	87	0.85	1.5	250	0.4	J	0.25
DA1 DE	90	62	125	0.87	3.5	250	0.4	J	0.25
DA4	95	120	140	1.6	16.0	400	1.1	J	0.45
DYMAX 1	108	82	168	1.6	4.0	300	0.6	A	2.20 (12 v)
DYMAX 2	127	82	168	2.0	3.0	300	0.6	B'	0.24
DYMAX 2A	127	82	168	2.0	1.5	150	0.3	B	0.24
CAPEX 1D	143	86	149	1.6	15.0	400	0.9	A	3.0 (12 v)
CAPEX 2D	142	92	139	2.0	6.0	400	0.9	B	0.25
CAPEX 2DC	160	180	230	3.7	6.0	400	0.9	B	0.25
M361	157	111	168	3.6	16.0	530	1.5	C	0.3
M361C	180	210	280	6.1	16.0	530	1.5	C	0.3
M371	157	108	194	3.8	16.0	530	1.5	F	4.5 (12 v)
F85 DE	192	120	280	6.8	30.0	*650	1.8	D	1.0
F85DEC	235	230	320	9.0	30.0	480	1.8	D	1.0
F85 DEP	240	120	280	7.0	30.0	*650	1.8	D	1.0
F85 DER	192	120	280	7.0	20.0	*700	4.0	D	1.0
R85 SE	190	143	273	6.5	18.0	500	1.6	C H	0.4
DUPLEX 22	127	220	232	8.0	40.0	600	2.0	D	1.0
DUPLEX 44	210	225	315	13.1	70.0	*740	2.0	C	1.0

*The vacuum figure stated is with the pump heads connected in series.



Model F85 DE
(Charles Austen Pumps Ltd)

Bristol Pneumatic Ltd

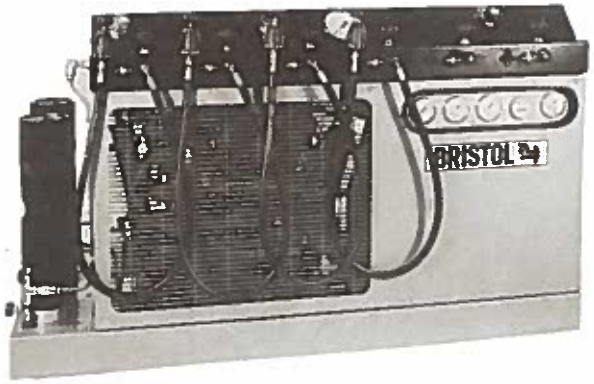
Details of all models produced are summarized in the following tables.

TRACTOR MOUNTING MODELS

Model	Displacement ft^3/min	Displacement m^3/min	FAD ft^3/min	FAD m^3/min	Max Pressure lb/in^2	Max Pressure bar	Speed rev/min	Mounting
TD140	162	4.58	140	3.96	100	7	1400	Ford tractor
IH145	172	4.86	145	4.10	100	7	1500	International Harvester tractor
MF140	161	4.58	140	3.96	100	7	1406	Massey Ferguson tractor
3PL140	165	4.67	140	3.96	100	7	1393	Universal mounting

LIGHT INDUSTRIAL RANGE

Model	Displacement ft^3/min	Displacement m^3/min	FAD ft^3/min	FAD m^3/min	Max Pressure lb/in^2	Max Pressure bar	Motor kW	Air Receiver ft^3	Air Receiver gals	Air Receiver litres
307/KL6	6	0.167	4	0.11	150	10.34	1.1	4.0	25.2	115
307/K10	10	0.283	7	0.20	200	13.78	2.2	4.0	25.2	115
307/B15	15	0.425	12	0.34	200	13.78	3.0	7.9	49.2	222
307/B20	20	0.566	15	0.42	200	13.78	4.0	7.9	49.2	222
307/D30	30	0.850	23	0.64	200	13.78	5.5	7.9	49.2	222
307/D40	40	1.130	30	0.84	200	13.78	7.5	7.9	49.2	222
309/KL6	6	0.167	4	0.11	150	10.34	1.1	4.0	25.2	115
309/K10	10	0.283	7	0.20	200	13.78	2.2	4.0	25.2	115



Bristol model BP18 charging set.

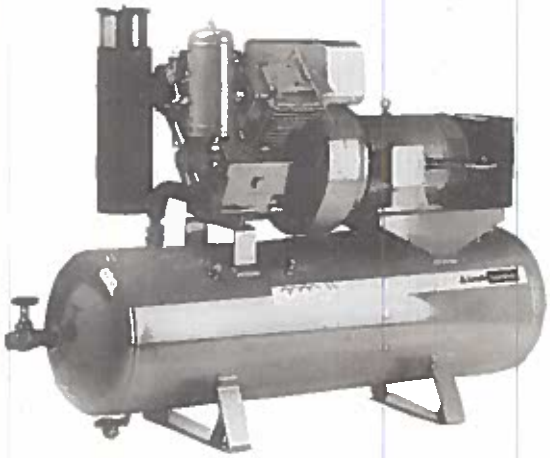
HIGH PRESSURE COMPRESSORS

Model	Displacement		FAD		Max Pressure		Motor
	ft^3/min	m^3/min	ft^3/min	m^3/min	lb/in^2	bar	hp
308/KL6H	6	0.170	4.25	0.120	450	31.5	2.0
308/A11H	11	0.311	7.5	0.212	400	28	4.0
308/C18H	18	0.510	12.0	0.340	300	21	7.5
308/BM3S16	16	0.453	12.0	0.340	500	35	7.5
308/BM3S20	20	0.566	15.5	0.439	500	35	7.5
308/BM3S24	24	0.679	20.0	0.566	500	35	10.0
BP2.4-40	4	0.114	2.0	0.057	4000	276	3.0
BP2.4-45	4	0.114	2.0	0.057	4500	310	3.0
BP5-40	7	0.198	5.0	0.141	4000	276	5.5
BP5-50	7	0.198	5.0	0.141	5000	345	5.5
BP12-40	16	0.453	11.0	0.312	4000	276	10.0
BP12-50	16	0.453	10.5	0.311	5000	345	15.0
BP18-40	23	0.651	18.0	0.509	4000	276	15.0
BP18-50	23	0.651	18.0	0.509	5000	345	15.0

BroomWade (CompAir Industrial Limited)

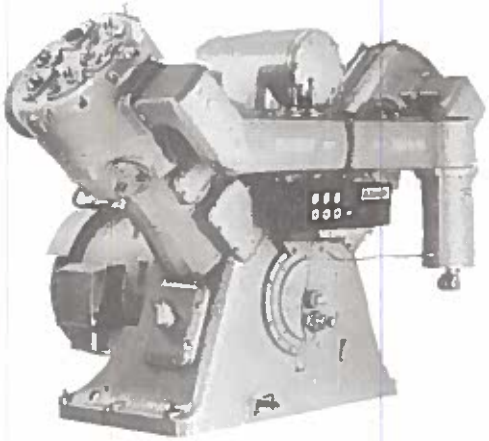
The BroomWade brand includes reciprocating and screw compressors ranging from ½ horsepower portable units to two-stage reciprocating machines of over 300 horsepower.

2000 Series compressors are designed for smaller compressed air needs and are portable units mounted on a horizontal receiver. The extensive range offers capacities from 1.2—40 lit/sec (2.6—85 ft^3/min) and working pressures from 5.2—13.8 bar (75—200 lb/in^2).



BroomWade 2000 series compressor.

BroomWade 600 series compressor.



BroomWade VMA 1000

6000 Series rotary screw compressors are of the oil-injected type and are produced in open or completely packaged form in an acoustically insulated enclosure including an instrument panel with comprehensive controls, instrumentation and fault indication. Smaller machines are air cooled, and the larger machines water-cooled. Capacities range from 35–700 lit/sec (75–148 ft^3/min), with working pressures from 7–10 bar.

V-form reciprocating compressors are produced in various ranges – V-Major, V-Compact and V-Master – in both lubricated and oil-free forms. All feature extremely fine balancing.

The *V-Compact* range offers capacities from 37–299 lit/sec (78.5–634 ft^3/min) and pressures from 1.5–10.5 bar (depending on number of stages.

V-Major compressors offer capacities from 225–1 530 lit/sec and pressures up to 20.5 bar (two-stage machines).

V-Master compressors cover capacities from 128–304 m^3/hr (76–180 ft^3/min), and working pressures from 10–350 bar.

Bryan Donkin Co Ltd

The Bryan Donkin range of reciprocating and turbo compressors encompasses single- and multi-stage machines suitable for a wide range of applications.

The reciprocating machines, including both vertical and horizontal opposed designs, in oil free, mini-lube and lubricated forms offer capacities up to 26 700 Nm^3/h, (10^6 scfh) and pressures up to 492 bar (7 000 lb/in^2) in 1, 2, 4 and 6 throw crank configurations. These high pressure machines lend themselves well to high pressure gas storage systems and numerous applications in the chemical, petro-chemical and process industries.

Bryan Donkin manufacture a comprehensive range of single-stage centrifugal SD and SG type blowers suitable for a wide variety of applications such as pneumatic conveying, sewage aeration and any industrial process where large volumes of air need to be handled at medium pressure *viz*. Volume flows up to 20 m^3/sec (42 000 scfm) with 700 mbar (10 lbf/in^2) pressure rise. One of the features of this range of machines is that inlet guide vane control can be fitted as standard to provide volume flow control with a constant speed motor without recourse to a variable speed prime mover. This type of control provides facilities for manual or most forms of automatic control to deal efficiently and economically with variations in throughput or discharge pressure.

The Bryan Donkin Mk II Roots type compressor/exhauster is an improved range of positive displacement machines. Operation is both simple and effective, two rotors mounted on parallel shafts within a casing rotate in opposite directions through a pair of timing gears. The profile and machine finish of the rotors is such that accurate clearance is maintained between the two, and between the rotors and casing. The Mk II Roots type blowers are designed to give a pressure rise of 1.03 bar from atmosphere on all sizes of machine with a capacity range of 0.28 m^3/s, to 9.5 m^3/s, (592 scfm to 18 000 scfm).

As an exhauster vacuums up to 380 mm Hg can be achieved under normal dry operation and 560 mm Hg with water injection to the inlet side.

These machines are suitable for sewage aeration, pneumatic handling, furnace air blowing, gas boosting, coke oven gas exhausting, steam boosting and circulation for de-salination plant and vacuum pump applications.

CompAir Industrial Ltd

CompAir Industrial are the largest UK manufacturers of compressed air equipment, group products including *BroomWade* stationary and portable compressors; *Reavell* high and low pressure stationary compressors; *Holman* portable compressors; *Hydrovane* portable and stationary compressors; and *Kellogg-American* stationary compressors. For details, see under individual company names.

Consolidated Pneumatic Tool Co Ltd.

Details of the range of single screw 'CP' portable compressors manufactured by this company are as follows:

Model	140	185	250	385
speed rev/min	2350	2100	2350	2450
capacity ft^3/min	140	185	250	385
m^3/min	4	5.2	7	10.9
normal working pressure lb/in^2	100	100	100	100
bar	7	7	7	7
maximum working pressure lb/in^2	125	125	125	125
bar	8.5	8.5	8.5	8.5
maximum operational ambient °F	125	125	125	125
°C	52	52	52	52
engine	Ford 2722E Perkins 4.236	Ford 2722E Perkins 4.236	Ford 2723E Perkins 6.3544	6M3/53T
noise level (silenced model) dB(A) @ 7M	74	74	74	75

The industrial range of single/screw compressors consists of six sizes in a standard range (125 lb/in^2 max) and an HP range (145 lb/in^2 max). Both ranges are offered in the air cooled or water cooled versions. With the addition of the CP High Efficiency feature, they are also offered for 'oil free' duties.

CP 140 single/screw compressor

Single/Screw CP Industrial Stationary Compressors

Model		140	175	210	265	340	445
Speed	rev/min	2950	2950	2950	2950	2950	2950
Actual delivery	ft^3/min	141	178	211	275	360	460
	m^3/min	4	5	6	7.9	10.2	13
Normal working pressure	lb/in^2	100	100	100	100	100	100
	bar	7	7	7	7	7	7
Maximum working pressure (standard)	lb/in^2	125	125	125	125	125	125
	bar	8.5	8.5	8.5	8.5	8.5	8.5
Maximum working pressure (hp range)	lb/in^2	145	145	145	145	145	145
	bar	10	10	10	10	10	10
Motor	kW	30	37	45	56	75	93
	hp	40	50	60	75	100	125
Power consumed (at 100 lb/in^2 discharge)	kW	27.4	33.9	36.9	49.6	59.3	77.3
	hp	36.7	45.4	49.5	66.5	79.5	103.6
Outlet discharge temp (over ambient)	°F	14–18	14–18	14–18	14–18	14–18	24–18
	°C	8–10	8–10	8–10	8–10	8–10	8–10

Dawson, McDonald & Dawson Ltd

This company produces the 'Compton' range of oil-free compressors, gas circulating pumps and vacuum pumps.

Compton type D/351 VM

The range starts with the smallest unit the D/351VM-MX2, powered by an open construction shaded-pole motor — this pump has a displacement of 0.65 ft^3/min (17.5 litres) and progresses via an extremely popular machine the D/351VM — which finds many applications in uses as diverse as photocopying machines, agricultural chemical applications, record cleaning machines, liquid level monitors, *etc* — to considerably larger units.

These larger units having single or multiple heads in addition to their uses as air compressors are widely used for gas circulation duties in chemical plants, nuclear research power stations, *etc*, as they can be built to a high standard of gas tightness having no glands or shaft seals.

Gardner-Denver (Cooper Air Compressor Division)

This (American) company produces the Electra-Saver compressors which is claimed to save up to 25% per year in energy costs over any comparable sized screw compressor. Electra-Saver compressors range in size from 100–500 horsepower (75–375 kW).

Gast Manufacturing Co Ltd

Gast Manufacturing Company manufacture a range of precision built oil free piston compressors with a continuous pressure rating of 7 bar (100 lb/in^2) and air flows to 4.8 lit/sec (10.2 ft^3/min). All incorporate high starting torque integral motors for single or three phase supplies in the power range from 1/6 to 1½ hp (0.12 to 1.12 kW).

Piston rings and skirts are of self-lubricating Teflon (a DuPont material), motor bearings are greased sealed for life and cylinder heads and pistons are of cast aluminium with hard coated aluminium cylinders. Other features include a special design of lap-joint piston ring with stainless steel backing seal and corrosion resistant sandwich type valves. All units are available on a number of alternative sizes of horizontal and vertical air receivers.

Diaphragm Models — these are available as single- or double-diaphragm units with pressure ratings of 4 bar (60 lb/in^2) and free air flows up to 1½ lit/sec (3.2 ft^3/min). Diaphragms are of a special high temperature synthetic fabric coated with a hydrocarbon based elastomer and a special feature of all models is the very low noise level (in the mid 50 dB(A) range).

Rotary Vane Models — these are essentially low pressure units providing a maximum pressure of 1.25 bar (15/20 lb/in^2). Both oil free and lubricated models are available up to 26 lit/sec (55 ft^3/min).

Arthur Gilderson Ltd

The Robay range of compressors marketed by this company are heavy duty two-stage reciprocating machines. Models are available from 2.25–80 ft^3/min (64–2 265 lit/min) either base-plate or receiver mounted. Standard pressure ranges are 100–125 lb/in^2 (7.8–9.5 bar) and 150–175 lb/in^2 gauge (10–12 bar). Other models are available up to 250 lb/in^2 (17 bar).

Howden Compressors

Following is a summary of the range of oil free, refrigeration, lubricated screw and Godfrey blowers manufactured by this company.

Model	Type	Installation	Capacity ft^3/min (m^3/hr)	Discharge Pressure lb/in^2 (bar)	Input HP	Application
Single-stage range	Rotary oil free wcrew	Stationary or skid mounted	500–11000 (850–18700)	Up to 250 (17.24)	117–3200	Industrial air and petrochemical gases
Booster range	Rotary oil free screw	Stationary or skid mounted	600–11000 (1020–18700)	Up to 250 (17.24)	100–5000	Industrial air and petrochemical gases
Two-stage	Rotary oil free screw	Stationary or skid mounted	1500–11000 (2550–18700)	Up to 250 (17.24)	571–5000	Industrial air and petrochemical gases
Single-stage oil injected gas/air/ refrigeration range	Rotary screw capacity control	Stationary or skid mounted	200–5200 (340–8840)	350 (24.14)	Up to 5000	Industrial air, petrochemical gases and refrigerant gases
Howdenair	Packaged lubricated screw air compressors	Stationary or skid mounted	65–1000	60–200 (4.14–13.8)	20–250	Industrial marine and instrument air systems
Godfrey blower range	Roots type positive displacement oil free	Stationary and vehicle-borne	30–1200	15 (1)	2–75	Pneumatic conveying bulk tankers, aeration, etc.

HPC Engineering Ltd

This company produces both reciprocating (piston) and screw air compressors, as detailed in the tables. Screw compressors are supplied mounted with all ancillaries fully enclosed in noise-reducing cabinets.

HPC PLUSAIR ROTARY SCREW AIR COMPRESSORS

	Air Delivered From Package in Accordance With ISO 1217		Maximum Pressure		Motor	
Model	m^3/min	ft^3/min	bar	lb/in^2	kW	hp
SK11	1.1	39	7.5	109	7.5	10
	0.9	32	10	145		
	0.7	25	13	188		
SK18	1.75	62	7.5	109	11	15
	1.5	53	10	145		
	1.15	41	13	188		
SK25	2.4	85	7.5	109	15	20
	2.1	73	10	145		
	1.7	59	13	188		
AS35	3.5	123	7.5	109	22	30
	3.1	108	10	145		
	2.7	94	13	188		
BS44	4.4	155	7.5	109	30	40
	4.0	141	10	145		
	3.5	124	13	188		
BS50	5.0	177	7.5	109	30	40
	4.29	152	10	145		
	3.74	132	13	188		
BS60	6.17	218	7.5	109	37	50
	5.25	185	10	145		
	4.6	162	13	188		
CS90	8.9	314	7.5	109	55	75
	7.75	274	10	145		
	6.68	236	13	188		
CS120	12.2	431	7.5	109	75	100
	10.25	362	10	145		
	8.9	314	13	188		
DS140	13.8	487	7.5	109	75	100
	11.6	410	10	145		
	9.0	318	13	188		
DS170	16.5	583	7.5	109	90	125
	14.2	501	10	145		
	11.4	403	13	188		
DS200	20.0	706	7.5	109	110	150
	17.3	611	10	145		
	14.3	505	13	188		
DS220	22.0	777	7.5	109	122	166
	19.0	671	10	145		
	15.7	554	13	188		
ES240	23.4	826	7.5	109	132	177
	20.5	724	10	145		
	17.4	614	13	188		
ES280	27.4	968	7.5	109	160	215
	24.3	858	10	145		
	20.8	735	13	188		
FS340	33.8	1165	7.5	109	200	268
	28.0	1024	10	145		
	23.5	883	13	188		
F420	42.0	1483	7.5	109	250	335
	35.5	1254	10	145		
	29.0	1024	13	188		

HPC PISTON RANGE — AIR COMPRESSORS

150 lb/in² MWP (10.3 bar) Electric Drive — Horizontal and Vertical Receiver Mounted

Model	Design	Piston Displacement ft³/min	Piston Displacement lit/min	Free Air Delivery ft³/min	Free Air Delivery lit/min	Motor hp	Motor kW	Receiver Capacity gall	Receiver Capacity litre
SV4	1-stage	3.2	90	2.0	56	0.5	0.37	11.0	50
SV6	1-stage	5.7	161	3.7	104	1.5	1.1	15.8	72
SH6	1-stage	5.7	161	3.7	104	1.5	1.1	25.6	116
SH10	2-stage	8.0	226	6.0	169	2.0	1.5	25.6	116
SH12	1-stage	14.3	405	8.7	246	3.0	2.2	48.0	218
SH15	1-stage	17.0	481	10.3	291	4.0	3.0	48.0	218
SH26	1-stage	27.4	776	18.6	526	5.5	4.0	48.0	218
SH42	2-stage	32.6	923	26.0	736	10.0	7.5	55.2	251
SH65	1-stage	67.1	1900	46.0	1302	15.0	11.0	82.3	374
SH105	2-stage	79.0	2237	61.0	1727	20.0	15.0	110	500

200 lb/in² MWP (13.8 bar) Electric Drive — Horizontal and Vertical Receiver Mounted

TSH9	2-stage	8.0	226	6.0	169	2.0	1.5	25.6	116
TSV9	2-stage	8.0	226	6.0	169	2.0	1.5	37.4	170
TSH13	2-stage	12.7	359	10.5	297	3.0	2.2	48.0	218
TSV13	2-stage	12.7	359	10.5	297	3.0	2.2	37.4	170
TSH15	2-stage	15.0	424	12.0	339	4.0	3.0	48.0	218
TSV15	2-stage	15.0	424	12.0	339	4.0	3.0	37.4	170
TSH20	2-stage	20.0	566	16.2	458	5.5	4.0	48.0	218
TSV20	2-stage	20.0	566	16.2	458	5.5	4.0	37.4	170
TSH32	2-stage	32.6	923	26.0	736	10.0	7.5	55.2	251
TSH75	2-stage	75.0	2124	58.0	1643	20.0	15.0	110	500

150 lb/in² MWP (10.3 bar) Petrol Drive — Horizontal Receiver Mounted

SH10GP	2-stage	8.0	226	6.0	169	5 petrol	25.6	116
SH15GPSR	1-stage	17.0	481	10.3	291	7 petrol	25.6	116
SH26GP	1-stage	27.4	776	18.6	526	10 petrol	55.2	251
SH26GPSR	1-stage	27.4	776	18.6	526	10 petrol	25.6	116
SH42GP	2-stage	32.6	923	26.0	736	14 petrol	55.2	251

150 lb/in² MWP (10.3 bar) Diesel Drive — Horizontal Receiver Mounted

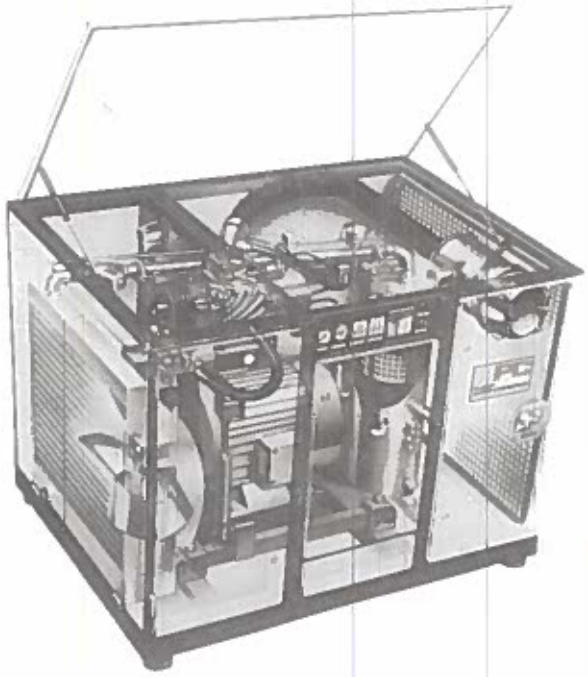
SH15GDSR	1-stage	17.0	4814	10.3	291	6 diesel	25.6	116
SH26GD	1-stage	27.4	776	18.6	526	6 diesel	55.2	251
SH42GD	2-stage	32.6	923	26.0	736	10 diesel	55.2	251

125 lb/in² MWP (8.6 bar) Portables and Mobiles

Professional	4.0	113	2.5	70	0.5 hp elec	—	No receiver
Portable 4	3.2	90	2.0	56	0.5 hp elec	2 hp pet	No receiver
Spray/Mobile 8	8.0	226	6.0	169	1.5 hp elec	5 hp pet	2 x 2 (2 x 9)
Mobile 15	17.0	481	10.3	291	3.0 hp elec	7 hp pet	2 x 2 (2 x 9)
Mobile 26E	27.4	776	18.6	526	5.5 hp elec	—	2 x 2 (2 x 9)
Mobile 26P	27.4	776	18.6	526	—	10 hp pet	2 x 2.7 (2 x 12.5)

150 lb/in² MWP (10.3 bar) Electric Drive — Base Mounted

Model	Design	Displacement ft³/min	Displacement lit/min	Air Delivered at 100 lb/in² ft³/min	Air Delivered at 100 lb/in² lit/min	Air Delivered at 150 lb/in² ft³/min	Air Delivered at 150 lb/in² lit/min	Motor kW	Motor hp
LBU20	2-stage	79	2237	61	1728	57	1614	15	20
LBU30	2-stage	109	3087	80	2266	74	2096	22	30
LBU40	2-stage	158	4475	112	3172	108	3059	30	40
LBU50	2-stage	213	6032	155	4390	153	4333	37	50



Typical HPC Plusair rotary screw compressor. The 18 models range from 10 hp delivering 39 cfm at 7.5 bar to 335 hp delivering 1 483 cfm at 7.5 bar. Maximum working pressures up to 200 lb/in^2 are available.

The Hydrovane Compressor Co Ltd

Hydrovane compressors are highly developed oil-sealed rotary vane machines, produced as packaged units together with electric motor, stand, pre-wired starter and integral control system. Standard packaged units can also be supplied fitted with an aftercooler and auto moisture trap. Types and performance details as per table.

Hydrovane 12 PV compressor.

HYDROVANE INDUSTRIAL COMPRESSORS

Model	Free Air Delivered l/s	Free Air Delivered ft^3/m	Working Pressure bar	Electric Motor	Noise Level at 1 metre
6 PUA	2.4	5.2	7/10	1.1kW	68 dB(A)
12 PUA	4.7	10.0	7/10	2.2 kW	65 dB(A)
13 PUA	5.3	11.4	7/10	2.2 kW	70 dB(A)
23 PUA	10.1	21.5	7/10	4.1 kW	74 dB(A)
33 PUA	14.1	30	7/10	5.5 kW	75 dB(A)
43 PUA	18.4	39	7/10	7.5 kW	76 dB(A)
66 PUA	29.5	55	7	11.2 kW	76 dB(A)
128 PUA	45.7	97	8	18.5 kW	80 dB(A)
170 PU	70.8	150	7	30 kW	89 dB(A)
250 PU	96.7	205	7	45 kW	88 dB(A)
350 PU	141.7	300	7	56 kW	88 dB(A)
400 PU	175.0	371	7	75 kW	89 dB(A)

Ingersoll-Rand Co Ltd

Examples of portable compressor units produced by this company include:-

Model DR85 2 430 l/m at 7 bar (85 ft^3/m at 100 lb/in^2)
With Perkins 3—152 diesel engine and two wheel running gear for speed towing.
With Ford 2504E diesel engine Simms spring starter, two wheel running gear for speed towing.

Model DRAF 125 3 570 l/m at 7 bar (125 ft^3/m at 100 lb/in^2)
With Ford 2711E diesel engine, two or four wheel running gear for speed towing.

Model DRF 175 5 000 l/m at 7 bar (175 ft^3/m at 100 lb/in^2)
With Ford 2711E diesel engine and two wheel running gear for speed towing.
This unit also supplied less running gear.

Model DRF 250 7 000 l/m at 7 bar (250 ft^3/m at 100 lb/in^2)
With Ford 2714E diesel engine and four wheel running gear for speed towing.
This unit also supplied with two wheel running gear for speed towing or less running gear suitable for truck mounting.

Model DR 365 10 400 l/m at 7 bar (365 ft^3/m at 100 lb/in^2)
With Cummins V 504 or Rolls Royce C4FNL diesel engines and mounted on four pneumatic-tyred wheels.
The above also supplied less running gear suitable for truck mounting.

Model DRCH 480 13 700 l/m at 7 bar (480 ft^3/m at 100 lb/in^2)
With Cummins NH220-Bi or Rolls Royce C6FNL diesel engines mounted on four pneumatic-tyred wheels.

Model DR 600 17 000 l/m at 7 bar (600 ft^3/m at 100 lb/in^2)
With Cummins NH220-Bi or Rolls Royce C6FNL diesel engines mounted on four pneumatic-tyred wheels.
The above also supplied less running gear suitable for truck mounting.

TYPE 40 COMPRESSORS

Standard two-stage type 40 compressors for operation at pressures up to 7 bar (100 lb/in^2). Assembled packaged units complete with motor and receiver.

Model	Free Air Delivered at 7 bar (100 lb/in^2) l/min (cu ft/min)	Speed rev/min	Motor hp
25	3623 (128)	970	30
40H	5663 (201)	970	50
50	7248 (256)	970	60
75H	11326 (401)	970	96
100	14158 (509)	970	120

Joy Manufacturing Co (UK) Ltd

The complete range of Joy compressors covers single-stage oil-injected rotary screw machines (Twistair); lubricated and oil free reciprocating compressors (WG and WN series) and centrifugal machines (Turbo-Air and MSG) of between two and six stages.

The Joy WG series are vertical single-stage, single cylinder, double-acting, water-cooled units designed for V belt drive. The main frames are designed for 5 in, 7 in or 9 in stroke. Operating pressures vary from 1 to 9 bar (15 to 130 lb/in^2) depending upon capacity. Capacity range (6 models) 2.3–9.6 m^3/min (80–340 ft^3/min). They may be used as vacuum pumps and are also suitable for handling some gases. WGOL models have dry cylinder head construction and deliver oil-free air. Capacity range (6 models) 2.4–9.9 m^3/min (85–350 ft^3/min).

The Joy WN series are V configuration, two-stage, two cylinder, double-acting, water-cooled units with a highly efficient water-cooled heat exchanger fitted between the first and second stages. They are designed for flange mounted electric motor drive but may also be utilized with direct coupled or belt drives. Operating pressures vary from 4.2–105 bar (60–150 lb/in^2) according to capacity. Capacity range (3 models) 11.5–30.3 m^3/min (415–1 070 ft^3/in). WNOL models feature the Joy dry cylinder construction, using stainless steel cylinder liners, the T-block design of piston with rings in specially formulated PTFE, an extra length piston rod and special seals, to ensure oil free air delivered. Capacity range (3 models) 12.0–13.6 m^3/min (475–1 115 ft^3/min).

'Turbo-Air' and MSG centrifugal compressors are a range of centrifugal compressor packages, compiled from matched impellers, pinions, scrolls and diffusers and built into standard compressor frames. The design is compact and allows for the mounting on a common frame of all the essential items – compressor, intercoolers, lubrication system, driver and coupling and the control panel.

The whole forms a 'package' of 25 models for a wide range of plant and process air duties within a span of 2 000–66 300 m^3/hr at pressures ranging from 1.4 to above 52 bar (1 200–39 000 ft^3/min at 20 to above 750 lb/in^2). Delivered air is completely oil-free.

Twistair screw compressors are a range of packaged compressors which utilize an oil-injected rotary screw unit with electric motor drive. All machines are air-cooled and supplied complete and ready for start-up without requiring foundations. Capacity range (18 models) 2.6–21 m^3/min (90–740 ft^3/min). Models are available with or without sound attenuated cabinets, according to the needs of the installation.

Lacy-Hulbert & Co Ltd

Lacy-Hulbert produce a comprehensive range of rotary and reciprocating vacuum pumps and compressors having displacements from 0.9–240 ft^3/min.

Units can be of the oil free or lubricated type. In addition, units are also available for centralized vacuum or compressed air systems for use in either industrial or medical fields.

Range	General Description	Model	Displacement m^3/hr	Displacement ft^3/min	Pressure lb/in^2	Vacuum in Hg
R	Sliding vane type air compressor/ vacuum pump: oil free or lubricated.	R1.5	1.5	0.9	15	25
		R3	3	1.75	15	25
		R5	5	2.9	15	25
		R7	7	4.1	15	25
		R10	10	5.9	15	25
		R16	16	9.4	15	25
		R25	25	14.7	15	25
		R40	40	23.5	15	25
RB	Sliding vane type vacuum pumps and air compressors: oil-free or lubricated.	RB1-5	9.01	5.3	10	24
		RB1-6	12.75	7.5	10	24
		RB2.00	14.45	8.5	10	25
		RB2.5	20.40	12.0	15	26
		RB2.8	28.90	17.0	15	26
		RB3	41.65	24.5	10	24
		RB4	59.50	35.0	10	24
		RB5	76.50	45.0	10	24
		RB6	88.40	52.0	10	24
FL	Sliding vane type vacuum pump: oil flooded.	FL10	17.0	10.0	–	29.5
		FL25	42.5	25.0	–	29.5
		FL60	102.0	60.0	–	29.5
		FL120	204.0	120.0	–	29.5
		FL180	306.0	180.0	–	29.5
		FL240	408.0	240.0	–	29.5
XB	Reciprocating type vacuum pumps: single stage.	XB1	42.5	25.0	–	28.5
		XB2	68.0	40.0	–	28.5
		XB3	115.6	68.0	–	28.5
		XB4	207.4	122.0	–	28.5
YB	Reciprocating type compressors: single stage.	YB1	20.4	12.0	50	–
		YB2	42.5	25.0	50	–
		YB3	59.5	35.0	50	–
		YB4	85.0	50.0	50	–

The Nash Engineering Company

Nash compressors and vacuum pumps are of the liquid ring type. Type CL compressors are high volume, low pressure machines available in 10 sizes giving 100–10 000 ft^3/min. Double lobe compressors are available in 28 basic models, from 10–2000 ft^3/min and 10–110 lb/in^2 pressure. Nash instrument air compressors cover the range 29–395 ft^3/min at pressures from 50–125 lb/in^2.

Double-lobe compressor.
(The Nash Engineering Co).

Nash vacuum pumps are available in 11 sizes in single-stage units from 170–13 330 ft^3/min; and 17 sizes in multi-stage units for vacua down to 7.62 mm.

Northey Compressors Ltd

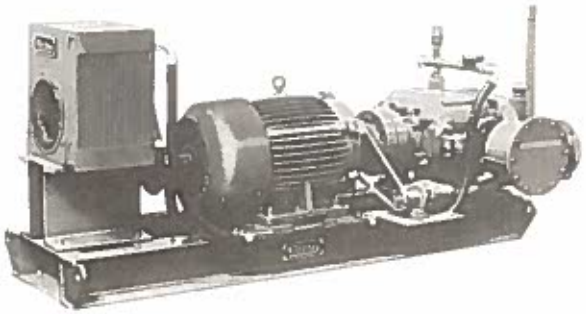
Northey specialize in the manufacture of rotary oil free air and gas compressors, boosters and vacuum pumps.

Northey single-stage 'oil free' compressors are available in air-cooled and water-cooled versions. Maximum delivery pressures for continuous operation are 1 bar gauge and 3 bar gauge water-cooled.

A range of two-stage water-cooled compressors is available for continuous operation at pressures of 4 bar gauge (7 bar gauge smaller models). Many standard Northey compressors are used as gas boosters and a specially designed range of boosters is also available.

'Oil free' and lubricated vacuum pumps are available with single-stage and two-stage models. Maximum vacuum obtainable 25 torr single-stage, 0.5 torr two-stage.

Latest developments include water-cooled models with completely self-contained water-cooling circuit using an air blast radiator and water circulation pump.



Typical Northey oil-free compressor.

Reavell (CompAir Industrial Ltd)

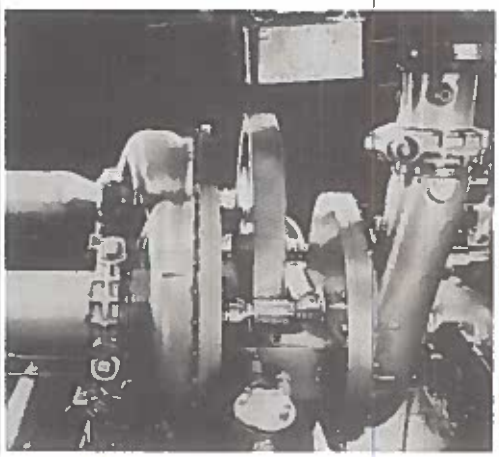
The Reavell range comprises reciprocating and rotary vane air and gas compressors as well as centrifugal blowers and exhausters. Details as per table.

REAVELL 5000 SERIES AIR-COOLED COMPRESSORS

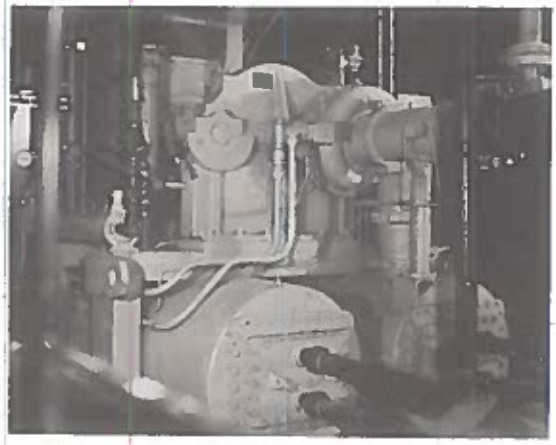
Model	Volume		Charging Rate		Max Pressure	Power	Speed
	m^3/h	ft^3/min	m^3/h	ft^3/min	bar	kW	r/min
5402A	2.92	1.72	3.5	2.05	275	2.2	2950
5404A	6.7	3.95	9.5	5.6	345	4	1300
5405A	8.6	5.05	12.0	7.05	345	5.5	1300
5406A	13.6	8.0	17.8	10.5	345	7.5	1800
5408A	28.9	17	37.4	22	345	15	1800
5409A	34.8	20.5	45	26.5	345	18.5	1800

United Technologies Elliott

Elliott PAP PLUS oil free compressors are centrifugal machines, available in sizes from 1 200–27 000 m^3/h (700–16 000 ft^3/min); and from 150–2 600 kW.



Elliott's PAP PLUS Plant use air package horizontally split construction for easy access to major components to encourage preventive maintenance.



Williams & James Ltd

Series 170 compressors are a heavy duty range suitable for either air or gas duties, pressure range is 28–280 bar (400–6000 lb/in^2) and capacities 9.4–19 dm^3/s (20–40 ft^3/min). Model 175 is a vee-form four-cylinder three-stage compressor giving 16.5 dm^3/s (35 ft^3/min) at 262 bar (3800 lb/in^2).

Series 200 compressors are air-cooled multi-stage machines, available with optional heavy-duty filtration systems with pressures up to 240 bar (6000 lb/in^2). Capacity is 5.7 dm^3/s (12 ft^3/min.

Series 140/150 compressors offer pressures up to 8.6 bar (125 lb/in^2) with a capacity of 66 dm^3/s (140 ft^3/min). The 140 series are air-blast cooled with integral aftercoolers. Model 156 is a water-cooled four-cylinder compressor capable of pressures up to 8.6 bar (125 lb/in^2) at outputs of 66 dm^3/s (140 ft^3/min).

Oil-free compressors are available in three ranges: Series 700 models, pressures up to 8.6 bar (125 lb/in^2), capacity 2.4–4.7 dm^3/s (5–10 ft^3/min); Series 190 models, pressures up to 8.6 bar (125 lb/in^2), capacity 9.4–56.6 dm^3/s (20–120 ft^3/min); and Series 600 models, pressures up to 10.3 bar (150 lb/in^2) and capacities 47.2–94.4 dm^3/s (100–200 ft^3/min). Model 695 gives 94.4 dm^3/s (200 ft^3/min) at pressures up to 10.3 bar (150 lb/in^2) and is complete with integral air-cooled interstage and aftercoolers.

SECTION 8

Buyers' Guide

Sub-Sections (a) Trade Names Index 459

(b) Classified Index of Pneumatic Equipment and Components Suppliers 462

(c) Alphabetical List of Manufacturers with addresses, telephone and telex numbers, of Head Office, Works and Branches 482

Sub-section (a)

TRADE NAMES INDEX

ALEMITE – Lubrication equipment – Stewart-Warner Ltd
AMLOC® II – Energy saving device for compressed air dryers – Pall Pneumatics Ltd
ANDEROL – Synthetic lubricants – Tenneco Colortrend B.V. – Anderol Div.
AQUADEX™ – Moisture indicator – Pall Pneumatics Ltd
ARO – Air tools and hoists – The Aro Corporation (UK) Ltd

BESTA – Air distribution equipment – Sperry Vickers Automation & Pneumatics
BIO–X – Compressed air sterilization filters – Domnick Hunter Filters Ltd.
'BOREAS' – Vacuum pumps and compressors – Lacy Hulbert & Co Ltd
BP ENERGOL – Compressor oils – BP Oils Ltd
BRISTOL DUPLEX – Air compressors – Bristol Pneumatic Ltd
BROOMWADE – Stationary and portable compressors, air treatment and ancillary products – CompAir Industrial Ltd

COMPTON – Oil-free diaphragm air compressor, oil-free diaphragm gas compressors and circulating pumps, oil-free vacuum pumps – Dawson,McDonald & Dawson Ltd

DECADE III™ – Heatless compressed air dryer – Pall Pneumatics Ltd
DYNA-QUIP – Ball valves, blow guns, quick release hose couplings – IMI Norgren Enots Ltd

ENOTS – Pneumatic cylinders and valves, tube and pipe fittings, tubing, pneumatic accessories, and filler caps – IMI Norgren Enots Ltd
EP7000 – Energy-saving fittings for pneumatic automation – Legris Ltd.

FESTO LTD – Pneumatic cylinders, valves, control systems – Festo Ltd
FIAM UTENSILI PNEUMATICI SpA – Pneumatic screwdrivers, drills, tappers, grinders, sanders, motors – Fiam Utensili Pneumatici SpA
FLUOSLIDE – Pneumatic control equipment, ie valves and cylinders – Midland Pneumatic Ltd.
FOXMELT – Adhesive systems – The Aro Corporation (UK) Ltd

GLOBE – Air motors – J.H.Fenner & Co Ltd

HEAT-LES® – Heatless compressed air dryer – Pall Pneumatics Ltd
'HOWDENAIR' – Packaged industrial compressors – Howden Compressors Ltd

ISOMAX – I.S.O. standard pneumatic valves – CompAir Maxam Ltd

JOUCOMATIC INTERNATIONAL – Pneumatic control equipment and process valves – Jouvenel et Cordier

LANG – Power pneumatic valves and cylinders – Sperry Vickers Automation & Pneumatics
LF3000 – Pneumatic push-in fitting system – Legris Ltd
LOADLEADER – Air hoists – The Aro Corporation (UK) Ltd
LUCIFER – Solenoid valves – Sperry Vickers Automation & Pneumatics

MISTCOOL – Liquid spraying system – IMI Norgren Enots Ltd
MODERNAIR – Valves and cylinders – The Aro Corporation (UK) Ltd
MSG – Centrifugal compressors – Joy Manufacturing Co (UK) Ltd

NORGREN – Compressed air equipment, fluidic devices, machine lubrication systems – IMI Norgren Enots Ltd
NORTHEY – Rotary clearance type 'oil free' air and gas compressors, boosters and vacuum pumps, lubricated vacuum pumps, railway vacuum brake exhausters – Northey Rotary Compressors Ltd.
'NUMATICS' – Air valves – Pnumatics Controls Ltd

OIL-X – Oil removing compressed air filters – Domnick Hunter Filters Ltd
OLYMPIAN – Compressed air processing equipment (plug-in) – IMI Norgren Enots Ltd

PALL® – Heaterless compressed air dryer – Pall Pneumatics Ltd
PARKER-HANNIFIN – Pneumatic cylinders, valves and air preparation equipment – Parker-Hannifin (UK) Ltd
PHARM AIR™ – Purification system – Pall Pneumatics Ltd
PISTONAIR – Compressed air motors – Joy Manufacturing Co (UK) Ltd
PNEUMAID – High pressure air logic system – CompAir Maxam Ltd
PNEUMETHODS – Pneumatic devices, sequencers, alarms, systems – Systematic Design & Development Ltd
POLYLOG – High pressure air integrated logic systems – CompAir Maxam Ltd
PORSILENT – Silencers – Sperry Vickers Automation & Pneumatics
PROVENAIR – Pneumatic cylinders – The Aro Corporation (UK) Ltd
PSIΨ – Ultra high efficiency filters, filter silencers, internal separators, activated carbon silencers – Process Scientific Innovations Ltd

RAY – Pressure snubbers – IMI Norgren Enots Ltd
REAVELL – High/low pressure stationary compressors, hydraulic controls – CompAir Industrial Ltd
RED-RING – Pneumatic cylinders and valves – Davis Pneumatic Systems Ltd

SCHUBERT & SALZER – Process control valves – Sperry Vickers Automation & Pneumatics
SPECKEN DRUMAG – Rotary actuators – Sperry Vickers Automation & Pneumatics
SYSTEM HF – High pressure hydraulic fitting system – Legris Ltd

THOR – Pneumatic tools – Stewart-Warner Ltd
THORITE – Air compressors – Thomas Wright (Bradford) Ltd
TRIQUAD – High pressure air/gas compressor – Williams & James (Engineers) Ltd
TURBO-AIR – Centrifugal compressors – Joy Manufacturing Co (UK) Ltd
TWISTAIR – Rotary screw compressors – Joy Manufacturing Co (UK) Ltd

ULTIPOR® – Filter medium – Pall Pneumatics Ltd
ULTRAIRE – High efficiency compressed air filters – IMI Norgren Enots Ltd

WILKERSON – Air services equipment – Sperry Vickers Automation & Pneumatics
W & J PNEUMERSTAT – Flow contoller, pressure reducer, filter – Williams & James (Engineers)

Sub-section (b)

CLASSIFIED INDEX OF PNEUMATIC EQUIPMENT AND COMPONENTS SUPPLIERS

Actuators, Rotary
Fluid Automation Systems SA

Aerators
Charles Austen Pumps Ltd

Aftercoolers, Air Cooled
Atlas Copco (GB) Ltd
CompAir Industrial Ltd
Ingersoll Rand Sales Co Ltd
NEI-APE Ltd Bellis & Morcom
Serck Heat Transfer
Thomas Wright (Bradford) Ltd

Aftercoolers, Water Cooled
Atlas Copco (GB) Ltd
CompAir Industrial Ltd
Ingersoll Rand Sales Co Ltd
Serck Heat Transfer
Thomas Wright (Bradford) Ltd

Agitators
The Aro Corporation (UK) Ltd

Air Actuators
CompAir Maxam Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Systematic Design & Developments Ltd
Thomas Wright (Bradford) Ltd

Air Conditioning Plant
Pall Pneumatics Ltd

Air Heaters
Domnick Hunter Filters Ltd

Alarm Warnings, Flow
IMI Norgren Enots Ltd (Shipston)
Systematic Design & Developments Ltd

Alarm Warnings, Level
Crouzet Ltd
Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Systematic Design & Developments Ltd

Alarm Warnings, Pressure
Crouzet Ltd
Davis Pneumatic Systems Ltd
Domnick Hunter Filters Ltd
Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Midland Pneumatic Ltd
R.G.S.Electro-Pneumatics Ltd
Systematic Design & Developments Ltd

Alarm Warnings, Temperature
Systematic Design & Developments Ltd

Alarm Warnings, Vacuum
Crouzet Ltd

Festo Ltd
Systematic Design & Developments Ltd

Bellows, Cylinder/Rod Protection
CompAir Maxam Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
R.G.S.Electro-Pneumatics Ltd
Sperry Vickers Automation & Pneumatics

Bits, Percussive
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll Rand Sales Co Ltd
Padley & Venables Ltd

Bits, Rotary
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll Rand Sales Co Ltd

Blowers, Diaphragm
Dawson, McDonald & Dawson Ltd

Blowers, Roots
Howden Compressors Ltd

Blowers, Screw
Howden Compressors Ltd
Ingersoll Rand Sales Co Ltd

Blowers, Sliding Vane
Lacy-Hulbert & Co Ltd

Blowers, Turbo
CompAir Industrial Ltd
Ingersoll Rand Sales Co Ltd

Boosters, Diaphragm
Dawson, McDonald & Dawson Ltd
Festo Ltd

Boosters, Piston
Atlas Copco (GB) Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
Ingersoll Rand Sales Co Ltd
NEI-APE Ltd Bellis & Morcom
R.G.S.Electro-Pneumatics Ltd
Williams & James (Engineers) Ltd

Boosters, Rotary
Atlas Copco (GB) Ltd
Howden Compressors Ltd
Ingersoll Rand Sales Co Ltd
Northey Rotary Compressors Ltd

Boosters, Turbo
CompAir Industrial Ltd

Bottles, Air: Tyre Inflating, Portable
Bristol Pneumatic Ltd

Brakes, Air Operated
Davis Pneumatic Systems Ltd

Breakers, Concrete and Paving
Atlas Copco (GB) Ltd
Consolidated Pneumatic Co

Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Padley & Venables Ltd
Williams & James (Engineers) Ltd

Breathing Air Sets
IMI Norgren Enots Ltd (Shipston)

Breathing Apparatus, Resuscitators
Domnick Hunter Filters Ltd
IMI Norgren Enots Ltd (Shipston)
Thomas Wright (Bradford) Ltd

Chucks, Vacuum (see also Air Chucks)
Charles Austen Pumps Ltd
Festo Ltd

Circulators, Air and Gas
Charles Austen Pumps Ltd
Northey Rotary Compressors Ltd

Clamps
Atlas Copco (GB) Ltd
Parker-Hannifin Ltd
Schrader Bellows Ltd

Compressor Oil Coolers, Air and Water Cooled
Serck Heat Transfer

Compressor Oils
BP Oil Ltd

Compressors, Air Diaphragm
Charles Austen Pumps Ltd
Dawson, McDonald & Dawson Ltd
Thomas Wright (Bradford) Ltd

Compressors, Air Reciprocating: Mobile
CompAir Industrial Ltd
HPC Engineering Ltd
Ingersoll Rand Sales Co Ltd
Williams & James (Engineers) Ltd
Thomas Wright (Bradford) Ltd

Compressors, Air Reciprocating: Portable
Charles Austen Pumps Ltd
CompAir Industrial Ltd
Dawson, McDonald & Dawson Ltd
Bristol Pneumatic Ltd
HPC Engineering Ltd
Ingersoll Rand Sales Co Ltd
Thomas Wright (Bradford) Ltd

Compressors, Air Reciprocating: Stationary or Fixed
Atlas Copco (GB) Ltd
NEI-APE Ltd Bellis & Morcom
Bristol Pneumatic Ltd
CompAir Industrial Ltd
Consolidated Pneumatic Tool Co Ltd
Dawson,McDonald & Dawson Ltd
HPC Engineering Ltd
Ingersoll Rand Sales Co Ltd
Joy Manufacturing Co (UK) Ltd
Lacy-Hulbert & Co Ltd
Ing Enea Mattei SpA
NEI-APE Ltd Bellis & Morcom
R.G.S.Electro-Pneumatics Ltd
Williams & James (Engineers) Ltd
Thomas Wright (Bradford) Ltd

Compressors, Air Reciprocating: Tractor Mounted
Bristol Pneumatic Ltd

Compressors, Air Rotary: Portable
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll Rand Sales Co Ltd
Ing Enea Mattei SpA

Compressors, Air Rotary: Roots (Lobed Impellers)
Howden Compressors Ltd
Ingersoll Rand Sales Co Ltd
Northey Rotary Compressors Ltd

Compressors, Air Rotary: Screw
Atlas Copco (GB) Ltd
CompAir Industrial Ltd
Howden Engineering Ltd
HPC Engineering Ltd
Ingersoll Rand Sales Co Ltd
Joy Manufacturing Co (UK) Ltd
Thomas Wright (Bradford) Ltd

Compressors, Air Rotary: Sliding Vane
CompAir Industrial Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll Rand Sales Co Ltd
Lacy-Hulbert & Co Ltd
Ing Enea Mattei SpA
Thomas Wright (Bradford) Ltd

Compressors, Air Rotary: Stationary or Fixed
Atlas Copco (GB) Ltd
CompAir Industrial Ltd
Consolidated Pneumatic Tool Co Ltd
Howden Compressors Ltd
HPC Engineering Ltd
Ingersoll-Rand Sales Co Ltd
Joy Manufacturing Co (UK) Ltd
Lacy-Hulbert & Co Ltd
Ing Enea Mattei SpA
Northey Rotary Compressors Ltd
Thomas Wright (Bradford) Ltd

Compressors, Air Rotary: Track Mounted
Ing Enea Mattei SpA

Compressors, Air Rotary: Vehicle Mounted
Consolidated Pneumatic Tool Co Ltd
Ing Enea Mattei SpA

Compressors, Air, Bottle Charging
Bristol Pneumatic Ltd
CompAir Industrial Ltd
Ingersoll Rand Sales Co Ltd
Williams & James (Engineers) Ltd

Compressors, Air Turbo: Centrifugal
CompAir Industrial Ltd
Ingersoll Rand Sales Co Ltd
Joy Manufacturing Co (UK) Ltd

Compressors, Dental
Atlas Copco (GB) Ltd
CompAir Industrial Ltd
HPC Engineering Ltd
Ingersoll Rand Sales Co Ltd
Williams & James (Engineers) Ltd
Thomas Wright (Bradford) Ltd

Compressors, Diving
Atlas Copco (GB) Ltd
Bristol Pneumatic Ltd
CompAir Industrial Ltd
Ingersoll Rand Sales Co Ltd
Williams & James (Engineers) Ltd

Compressors, Double Entry
Ingersoll Rand Sales Co Ltd

Compressors, Garage
Atlas Copco (GB) Ltd
Bristol Pneumatic Ltd
CompAir Industrial Ltd
Dawson,McDonald & Dawson Ltd
Howden Compressors Ltd
HPC Engineering Ltd
Ingersoll Rand Sales Co Ltd
Thomas Wright (Bradford) Ltd

Compressors, Gas
Atlas Copco (GB) Ltd
Charles Austen Pumps Ltd
NEI-APE Ltd Bellis & Morcom
Bristol Pneumatic Ltd
CompAir Industrial Ltd
Dawson,McDonald & Dawson Ltd

Howden Compressors Ltd
Ingersoll-Rand Sales Co Ltd
Joy Manufacturing Co (UK) Ltd
Northey Rotary Compressors Ltd
Williams & James (Engineers) Ltd

Compressors, Natural Gas (For Automotive Purposes)
Williams & James (Engineers) Ltd

Compressors, Oil Free: Diaphragm
Charles Austen Pumps Ltd
NEI-APE Ltd Bellis & Morcom
Dawson McDonald & Dawson Ltd
Thomas Wright (Bradford) Ltd

Compressors, Oil Free: Reciprocating
Atlas Copco (GB) Ltd
Charles Austen Pumps Ltd
CompAir Industrial Ltd
Ingersoll-Rand Sales Co Ltd
Joy Manufacturing Co (UK) Ltd
Williams & James (Engineers) Ltd
Thomas Wright (Bradford) Ltd

Compressors, Oil Free: Rotary
Atlas Copco (GB) Ltd
Howden Compressors Ltd
HPC Engineering Ltd
Ingersoll-Rand Sales Co Ltd
Lacy-Hulbert & Co Ltd
Northey Rotary Compressors Ltd

Compressors, Paint Spraying
Atlas Copco (GB) Ltd
CompAir Industrial Ltd
Dawson,McDonald & Dawson Ltd
Ingersoll-Rand Sales Co Ltd
Thomas Wright (Bradford) Ltd

Compressors, Portable
Atlas Copco (GB) Ltd
Charles Austen Pumps Ltd
CompAir Industrial Ltd
Dawson,McDonald & Dawson Ltd
HPC Engineering Ltd
Ingersoll-Rand Sales Co Ltd
Williams & James (Engineers) Ltd
Thomas Wright (Bradford) Ltd

Compressors, Refrigeration
Howden Engineering Ltd
Ingersoll-Rand Sales Co Ltd

Containers, Paint: Pressure
Thomas Wright (Bradford) Ltd

Controllers, Flow
Davis Pneumatic Systems Ltd
Drallim Controllers Ltd
Martonair Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Controllers, Level
The Aro Corporation (UK) Ltd
Crouzet Ltd
Festo Ltd
Schrader Bellows Ltd
Systematic Design & Developments Ltd

Controllers, Positional
Davis Pneumatic Systems Ltd
Martonair Ltd
R.G.S. Electro-Pneumatics Ltd
Schrader Bellows Ltd
Systematic Design & Developments Ltd

Controllers, Pressure
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Martonair Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Control Panels, Pneumatic Sequence
The Aro Corporation (UK) Ltd
CompAir Maxam Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Systematic Design & Developments Ltd

Counters, Pneumatic
Systematic Design & Developments Ltd

Couplings, Air Vacuum Line: Pipe
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Davis Pneumatic Systems Ltd
Drallim Fluid Components Ltd
Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Legris Ltd
Martonair Ltd
Parker-Hannifin (UK) Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics

Couplings, Air Vacuum Line: Quick Acting
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Davis Pneumatic Systems Ltd
Drallim Fluid Components Ltd
Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Jouvenel et Cordier
Legris Ltd
Martonair Ltd
Parker-Hannifn (UK) Ltd
Schrader Bellows Ltd

Couplings, Air Vacuum Line: Rotary
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Parker-Hannifin (UK) Ltd
Schrader Bellows Ltd

Couplings, Air Vacuum Line: Self-Sealing
Drallim Fluid Components Ltd
Festo Ltd
Jouvenel et Cordier
Martonair Ltd
Parker-Hannifin (UK) Ltd
Schrader Bellows Ltd

Couplings, Air Vacuum Line: Swivel
Parker-Hannifin (UK) Ltd
Schrader Bellows Ltd

Cutters, Rivet
Padley & Venables Ltd

Cutters, Tube
CompAir Maxam Ltd
Crouzet Ltd
Legris Ltd
Martonair Ltd
Padley & Venables Ltd

Cylinders, Air: Single-Acting
The Aro Corporation (UK) Ltd
CompAir Maxam Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
Fluid Automation Systems SA
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pneumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader-Bellows Ltd
Sperry Vickers Automation & Pneumatics
Systematic Design & Developments Ltd
Thomas Wright (Bradford) Ltd

Cylinders, Air: Double-Acting
The Aro Corporation (UK) Ltd
CompAir Maxam Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
Fluid Automation Systems SA
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pneumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Systematic Design & Developments Ltd
Thomas Wright (Bradford) Ltd

Cylinders, Air: Cushioned
The Aro Corporation (UK) Ltd
CompAir Maxam Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pneumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Systematic Design & Developments Ltd
Thomas Wright (Bradford) Ltd

Cylinders, Air: Duplex
CompAir Maxam Ltd

Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Pneumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Systematic Design & Developments Ltd

Cylinders, Air: Non-Cushioned

The Aro Corporation (UK) Ltd
CompAir Maxam Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pneumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Systematic Design & Developments Ltd
Thomas Wright (Bradford) Ltd

Cylinders, Air: Impact

Davis Pneumatic Systems Ltd
IMI Norgren Enots Ltd (Lichfield)
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Pneumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics

Cylinders, Air: Oscillating

Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Martonair Ltd
Mecman Ltd
Midland Pneumatics Ltd
Pneumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Systematic Design & Developments Ltd

Cylinders, Rotary

Festo Ltd
Martonair Ltd
Mecman Ltd
Pneumatics Controls Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics

Cylinders, Air: Telescopic

Davis Pneumatic Systems Ltd
IMI Norgren Enots Ltd (Lichfield)
Midland Pneumatic Ltd
Pneumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd

Cylinders, Air: Through Rod

CompAir Maxam Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pneumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Systematic Design & Developments Ltd

Cylinders, Air: Other Types

CompAir Maxam Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Pnumatics Controls Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Systematic Design & Developments Ltd

Cylinders, Magnetic

IMI Norgren Enots Ltd (Lichfield)

Dampers

Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Mecman Ltd
Schrader Bellows Ltd

Dehumidifiers, Air

Sperry Vickers Automation & Pneumatics
Ultrafilter Ltd

Desiccators
Domnick Hunter Filters Ltd
Ultrafilter Ltd

Diggers, Clay or Spade
Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd
Padley & Venables Ltd
Stewart-Warner Ltd

Drill Legs
Atlas Copco (GB) Ltd

Drill Rods
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Padley & Venables Ltd

Drills, Angle
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd
Thomas Wright (Bradford) Ltd

Drills, Auger
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd

Drills, Corner
Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd

Drills, Industrial: Adjustable Head
The Aro Corporation (UK) Ltd
Fiam Utensili Pneumatici SpA
Schrader Bellows Ltd

Drills, Industrial: Auto-Feed
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Schrader Bellows Ltd

Drills, Industrial: Corner
Fiam Utensili Pneumatici SpA
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd

Drills, Industrial: Fixed Angle Head
The Aro Corporation (UK) Ltd
Fiam Utensili Pneumatici SpA
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd

Drills, Industrial: Heavy Duty
Atlas Copco (GB) Ltd
Fiam Utensili Pneumatica SpA
Ingersoll-Rand Sales Co Ltd
Schrader Bellows Ltd
Stewart-Warner Ltd

Drills, Industrial: Light Duty
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd
Schrader Bellows Ltd
Stewart-Warner Ltd
Thomas Wright (Bradford) Ltd

Drills, Industrial: Medium Duty
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Fiam Utensili Pneumatici SpA
Ingersoll-Rand Sales Co Ltd
Schrader Bellows Ltd
Stewart-Warner Ltd
Thomas Wright (Bradford) Ltd

Drills, Industrial: Ultra-Lightweight
Fiam Utensili Pneumatici SpA
Ingersoll-Rand Sales Co Ltd

Drills, Industrial: Multi-Spindle
The Aro Corporation (UK) Ltd
Fiam Utensili Pneumatici SpA
Ingersoll-Rand Sales Co Ltd
Schrader Bellows Ltd
Stewart-Warner Ltd

Drills, Industrial: Self-Feed
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Schrader Bellows Ltd
Stewart-Warner Ltd
Thomas Wright (Bradford) Ltd

Drills, Wagon
Atlas Copco (GB) Ltd

Drills, Woodborers
Consolidated Pneumatic Tool Co Ltd
Schrader Bellows Ltd
Stewart-Warner Ltd

Drives and Prime Movers
Schrader Bellows Ltd

Dryers, Adsorption Heatless
Ultrafilter Ltd

Dryers, Adsorption Heat Regenerated
Ultrafilter Ltd

Dryers, Compressed Air and Gas
Pall Pneumatics Ltd

Dryers, Heaterless
Pall Pneumatics Ltd

Dryers, Heat Regenerated
Pall Pneumatics Ltd

Dryers, Refrigerant
Ultrafilter Ltd

Energy Saving Fittings
Legris Ltd

Ejectors, Power Press
Schrader Bellows Ltd

Exhausters (See also Vacuum Pumps)
CompAir Industrial Ltd
Lacy-Hulbert & Co Ltd
Northey Rotary Compressors Ltd

Filter Cartridges
Domnick Hunter Filters Ltd

Filter/Dryer, Systems
Utrafilter Ltd

Filters, Gas
Domnick Hunter Filters Ltd

Filters, High Pressure
Domnick Hunter Filters Ltd

Filters, Oil Removal
Ultrafilter Ltd

Filters, Sterilizing
Domnick Hunter Filters Ltd

Filters and Separators, Adsorption
CompAir Industrial Ltd
Domnick Hunter Filters Ltd
Pall Pneumatics Ltd
Process Scientific Innovations Ltd
Sperry Vickers Automation & Pneumatics
Williams & James (Engineers) Ltd

Filters and Separators, Air Intake
CompAir Industrial Ltd
Domnick Hunter Filters Ltd
Fiam Utensili Pneumatici SpA
Midland Pneumatic Ltd
Northey Rotary Compressors Ltd
Pall Pneumatics Ltd
Process Scientific Innovations Ltd
Watts Regulator (UK) Ltd

Filters and Separators, Air Pipeline (See also Desiccators)
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
CompAir Maxam Ltd
Crouzet Ltd
Domnick Hunter Filters Ltd
Fiam Utensili Pneumatici SpA
IMI Norgren Enots Ltd (Shipston)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Pall Pneumatics Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
Process Scientific Innovations Ltd
R.G.S. Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Filters and Separators, Vacuum
Domnick Hunter Filters Ltd
IMI Norgren Enots Ltd (Shipston)
Northey Rotary Compressors Ltd
Process Scientific Innovations Ltd

Flowmeters, Air
Drallim Controls Ltd
Systematic Design & Developments Ltd

Fittings, Air Line (See also Hose) : Ferrous
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
CompAir Maxam Ltd
Consolidated Pneumatic Tool Co Ltd
Davis Pneumatic Systems Ltd
Drallim Fluid Components Ltd
IMI Norgren Enots Ltd (Lichfield)
Legris Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
Process Scientific Innovations Ltd

R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Thomas Wright (Bradford) Ltd

Fittings, Air Line: Non-Ferrous

The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
CompAir Maxam Ltd
Consolidated Pneumatic Tool Co Ltd
Davis Pneumatic Systems Ltd
Drallim Fluid Components Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Legris Ltd
Martonair Ltd
Mecman Ltd
Pneumatics Controls Ltd
Process Scientific Innovations Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Fittings, Air Line: Plastic

Crouzet Ltd
Drallim Fluid Components Ltd
Festo Ltd
Martonair Ltd
Mecman Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics

Fittings, Air Line: Push-In

CompAir Maxam Ltd

Gauges, Pressure

The Aro Corporation (UK) Ltd
CompAir Maxam Ltd
Crouzet Ltd
Domnick Hunter Filters Ltd
Drallim Fluid Components Ltd
Festo Ltd
Jouvenel et Cordier
Lacy Hulbert & Co Ltd
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Northey Rotary Compressors Ltd
Parker-Hannifin (UK) Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Gauges, Vacuum

Crouzet Ltd
Lacy Hulbert & Co Ltd
Northey Rotary Compressors Ltd
Thomas Wright (Bradford) Ltd

Grinders

The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Fiam Utensili Pneumatici SpA
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd
Thomas Wright (Bradford) Ltd

Guns, Blow (or Dusters)

The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Ingersoll-Rand Sales Co Ltd
Jouvenel et Cordier
Midland Pneumatic Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Guns, Oiling

The Aro Corporation (UK) Ltd
Thomas Wright (Bradford) Ltd

Guns, Spray, Automatic

The Aro Corporation (UK) Ltd
Schrader Bellows Ltd
Stewart-Warner Ltd

Guns, Spray, Metal Coating

Stewart-Warner Ltd

Guns, Spray, Paint, Liquids

The Aro Corporation (UK) Ltd
Stewart-Warner Ltd

Hammers, Bush

Stewart-Warner Ltd

Hammers, Chipping and Caulking

The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd
Thomas Wright (Bradford) Ltd

Hammers, Forging

Stewart-Warner Ltd

Hammers, Jack
Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd

Hammers, Nail Driving
Ingersoll-Rand Sales Co Ltd

Hammers, Pick, Dry
Atlas Copco (GB) Ltd
Stewart-Warner Ltd

Hammers, Pick, Wet
Atlas Copco (GB) Ltd

Hammers, Riveting
Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd
Thomas Wright (Bradford) Ltd

Hammers, Scaling
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd
Thomas Wright (Bradford) Ltd

Hammers, Stone Working
Atlas Copco (GB) Ltd

Hoists, Air Motor Type
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd
Joy Manufacturing Co (UK) Ltd
Stewart-Warner Ltd
Thomas Wright (Bradford) Ltd

Hoists, Cylinder
Ingersoll Rand Sales Co Ltd
Midland Pneumatic Ltd
R.G.S.Electro-Pneumatics Ltd

Hoists, Overhead, Platform, Portable and Skip
Ingersoll Rand Sales Co Ltd

Horns, Pneumatic
Systematic Design & Developments Ltd

Hose
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Hose, Clips
Atlas Copco (GB) Ltd
Charles Austen Pumps Ltd
Consolidated Pneumatic Tool Co Ltd
Drallim Fluid Components Ltd
Festo Ltd
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Thomas Wright (Bradford) Ltd

Hose, Couplings
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Davis Pneumatic Systems Ltd
Drallim Fluid Components Ltd
Festo Ltd
Fiam Utensili Pneumatici SpA
IMI Norgren Enots Ltd (Shipston)
Legris Ltd
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Stewart-Warner Ltd
Thomas Wright (Bradford) Ltd

Hose, Fittings
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Davis Pneumatic Systems Ltd
Drallim Fluid Components Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Legris Ltd
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
R.G.S.Electro-Pneumatics Ltd

Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Hose, High Pressure

IMI Norgren Enots Ltd (Lichfield)
Parker-Hannifin (UK) Ltd
Thomas Wright (Bradford) Ltd

Hose, Low Pressure

Atlas Copco (GB) Ltd
Drallim Fluid Components Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Thomas Wright (Bradford) Ltd

Hose, Medium Pressure

Consolidated Pneumatic Tool Co Ltd
Drallim Fluid Components Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Martonair Ltd
Parker-Hannifin (UK) Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Thomas Wright (Bradford) Ltd

Hose, Plastic

Atlas Copco (GB) Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Drallim Fluid Components Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Hose, Reels

The Aro Corporation (UK) Ltd
IMI Norgren Enots Ltd (Lichfield)
Parker-Hannifin (UK) Ltd
Thomas Wright (Bradford) Ltd

Hose, Special Types

Davis Pneumatic Systems Ltd
Fluid Automation Systems SA

IMI Norgren Enots Ltd (Lichfield)
Parker-Hannifin (UK) Ltd
Schrader Bellows Ltd
Systematic Design & Developments Ltd

Hydraulic Check Units

Davis Pneumatic Systems Ltd
Festo Ltd
Fluid Automation Systems SA
Mecman Ltd

Indexing Tables, Rotary

Festo Ltd
Fluid Automation Systems SA
Jouvenel et Cordier
Schrader Bellows Ltd

Indicators, Pneumatic

Crouzet Ltd

Instruments

R.G.S Electro-Pneumatics Ltd

Intensifiers, Hydropneumatic

Davis Pneumatic Systems Ltd
Jouvenel et Cordier
Mecman Ltd
R.G.S Electro-Pneumatics Ltd
Schrader Bellows Ltd

Intensifiers, Pneumatic

CompAir Maxam Ltd
Davis Pneumatic Systems Ltd
Mecman Ltd

Intercoolers

CompAir Industrial Ltd
Serck Heat Transfer

Jacks, Pneumatic

Mecman Ltd

Logic Equipment, Air Operated

CompAir Maxam Ltd

Logic (Moving Part)

Crouzet Ltd

Lubricants, Oil

Tenneco Colortrend BV
BP Oil Ltd

Lubricants, Synthetic

Tenneco Colortrend BV

Lubricators
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
CompAir Maxam Ltd
Consolidated Pneumatic Tool Co Ltd
Crouzet Ltd
Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Jouvenel et Crodier
Mecman Ltd
Parker-Hannifin (UK) Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Stewart-Warner Ltd
Thomas Wright (Bradford) Ltd
Watts Regulator (UK) Ltd

Machines, Air Operated
Schrader Bellows Ltd
Stewart-Warner Ltd

Manometers
Mecman Ltd
Parker-Hannifin (UK) Ltd

Moisture Separators
(See also Filters & Separators)
The Aro Corporatin (UK) Ltd
Atlas Copco (GB) Ltd
CompAir Maxam Ltd
Consolidated Pneumatic Tool Co Ltd
Domnick Hunter Filters Ltd
Mecman Ltd
Process Scientific Innovations Ltd
Schrader Bellows Ltd
Thomas Wright (Bradford) Ltd

Motor Breakers
Atlas Copco (GB) Ltd

Motor Drills
Atlas Copco (GB) Ltd

Motors, Multi-Cylinder Piston Air Motors
Atlas Copco (GB) Ltd
J.H.Fenner & Co Ltd
Joy Manufacturing Co (UK) Ltd

Motors, Rotary Vane
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
J.H.Fenner & Co Ltd
Fiam Utensili Pneumatici SpA
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd
Thomas Wright (Bradford) Ltd

Motors, Turbine
The Aro Corporation (UK) Ltd
Ingersoll-Rand Sales Co Ltd

Nibblers and Shearing Machines
Atlas Copco (GB) Ltd
Fiam Utensili Pneumatici SpA

Nutrunners (See also Wrenches)
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Fiam Utensili Pneumatici SpA
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd
Thomas Wright (Bradford) Ltd

O-Rings
Festo Ltd

Packing Glands and Seals
Parker-Hannifin (UK) Ltd

Packings
Parker-Hannifin (UK) Ltd

Pipe Joints and Fittings
IMI Norgren Enots Ltd (Lichfield)
Legris Ltd
Parker-Hannifin (UK) Ltd
R.G.S.Electro-Pneumatics Ltd
Thomas Wright (Bradford) Ltd

Pipes, Ferrous
IMI Norgren Enots Ltd (Lichfield)
Legris Ltd
R.G.S.Electro-Pneumatics Ltd

Pipes, Flexible
Drallim Fluid Components Ltd
IMI Norgren Enots Ltd (Lichfield)
Legris Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Systematic Design & Developments Ltd
Thomas Wright (Bradford) Ltd

Pipes, Plastic
CompAir Maxam Ltd
Drallim Fluid Components Ltd
IMI Norgren Enots Ltd (Lichfield)
Legris Ltd

R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Pipes, Non-Ferrous

CompAir Maxam Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Pipework

R.G.S.Electro-Pneumatics Ltd
Systematic Design & Developments Ltd

Polishers

The Aro Corporation (UK) Ltd
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd

Pneumatic Conveying

Howden Compressors Ltd

Pneumatic Programmes

IMI Norgren Enots Ltd (Lichfield)

Presses, Impact

Martonair Ltd
Schrader Bellows Ltd

Presses, Squeeze

Schrader Bellows Ltd

Pumps (See also Vacuum Pumps) : Air Lift

CompAir Industrial Ltd
Lacy-Hulbert & Co Ltd
Stewart-Warner Ltd

Pumps, Displacement

The Aro Corporation (UK) Ltd
Ingersoll-Rand Sales Co Ltd

Pumps, Reciprocating

Ingersoll Rand Sales Co Ltd

Pumps, Sludge

Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd

Pumps, Sump

Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd

Rammers, Backfill

Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Padley & Venables Ltd
Stewart-Warner Ltd

Rammers, Foundry and Sand

Atlas Copco (GB) Ltd
Stewart-Warner Ltd

Rams

Festo Ltd
Martonair Ltd
Mecman Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Systematic Design & Developments Ltd

Receivers, Small

Atlas Copco (GB) Ltd
CompAir Industrial Ltd
Consolidated Pneumatic Tool Co Ltd
Festo Ltd
Martonair Ltd
Midland Pneumatic Ltd
R.G.S.Electro-Pneumatics Ltd
Systematic Design & Developments Ltd
Thomas Wright (Bradford) Ltd

Receivers, Medium

Atlas Copco (GB) Ltd
CompAir Industrial Ltd
Consolidated Pneumatic Tool Co Ltd
Festo Ltd
Midland Pneumatic Ltd
R.G.S Electro-Pneumatics Ltd
Thomas Wright (Bradford) Ltd

Receivers, Large

Atlas Copco (GB) Ltd
CompAir Industrial Ltd
R.G.S.Electro-Pneumatics Ltd

Regulators, Flow

Crouzet Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Legris Ltd
Martonair Ltd
Mecman Ltd

Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd
Watts Regulator (UK) Ltd

Regulators, Instrument

Davis Pneumatic Systems Ltd
IMI Norgren Enots Ltd (Shipston)
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd

Regulators, Liquid

The Aro Corporation (UK) Ltd
IMI Norgren Enots Ltd (Shipston)
Watts Regulators (UK) Ltd

Regulators, Pressure

The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
CompAir Maxam Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Jouvenel et Cordier
Legris Ltd
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Williams & James (Engineers) Ltd
Thomas Wright (Bradford) Ltd
Watts Regulator (UK) Ltd

Regulators, Not Self-Relieving

Davis Pneumatic Systems Ltd
IMI Norgren Enots Ltd (Shipston)
Jouvenel et Cordier
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Thomas Wright (Bradford) Ltd
Watts Regulator (UK) Ltd

Regulators, Remote Operated

Davis Pneumatic Systems Ltd
IMI Norgren Enots Ltd (Shipston)
Jouvenel et Cordier
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Watts Regulator (UK) Ltd

Regulators, Self-Relieving

Davis Pneumatic Systems Ltd
IMI Norgren Enots Ltd (Shipston)
Jouvenel et Cordier
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Thomas Wright (Bradford) Ltd

Regulators, Temperature

Pnumatics Controls Ltd

Regulators, Vacuum

Pnumatics Controls Ltd

Riveters, Aircraft

Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd

Riveters, Industrial

Atlas Copco (GB) Ltd
Fiam Utensili Pneumatici SpA
Ingersoll-Rand Sales Co Ltd

Rock Drills, Deephole

Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd

Rock Drills, Diamond

Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd

Rock Drills, Drifters

Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd

Rock Drills, Dustless (Wet)

Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd

Rock Drills, General Duty
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd

Rock Drills, Mobile
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd

Rock Drills, Pushers
Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd

Rock Drills, Self-Propelled
Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd

Rock Drills, Sinkers
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd

Rock Drills, Stopers
Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd

Sand Blast Plant
Atlas Copco (GB) Ltd

Sanders, Orbital
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Fiam Utensili Pneumatici SpA
Ingersoll-Rand Sales Co Ltd
Thomas Wright (Bradford) Ltd

Sanders, Reciprocating
Fiam Utensili Pneumatici SpA

Sanders, Rotary
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Fiam Utensili Pneumatici SpA
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd
Thomas Wright (Bradford) Ltd

Saws, Chain
Consolidated Pneumatic Tool Co Ltd

Saws, Circular
Atlas Copco (GB) Ltd
Fiam Utensili Pneumatici SpA
Ingersoll-Rand Sales Co Ltd

Saws, Reciprocating
Ingersoll-Rand Sales Co Ltd
Thomas Wright (Bradford) Ltd

Screwdrivers
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Fiam Utensili Pneumatici SpA
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd
Thomas Wright (Bradford) Ltd

Sensors, Gap, Leak and Proximity
Crouzet Ltd

Silencers, Compressor Delivery
IMI Norgren Enots Ltd (Shipston)

Silencers, Pneumatic Cylinders & Valves
CompAir Maxam Ltd
Domnick Hunter Filters Ltd
Festo Ltd
Fiam Utensili Pneumatici SpA
IMI Norgren Enots Ltd (Shipston)
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
Process Scientific Innovations Ltd
R.G.S. Electro-Pneumatics Ltd
Schrader-Bellows Ltd
Sperry Vickers Automation & Pneumatics

Silencers, Pneumatic Tool
Atlas Copco (GB) Ltd
IMI Norgren Enots Ltd (Lichfield)
Stewart-Warner Ltd

Snaps, Rivet
Atlas Copco (GB) Ltd

Solenoids
The Aro Corporation (UK) Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Mecman Ltd
Midland Pneumatic Ltd

Parker-Hannifin (UK) Ltd
Process Scientific Innovations Ltd
Schrader-Bellows Ltd
Sperry Vickers Automations & Pneumatics

Soot Blowers
Howden Engineering Ltd
Ingersoll-Rand Sales Co Ltd

Spike, Driver
Ingersoll-Rand Sales Co Ltd

Spray, Booths, Water-Washed
Atlas Copco (GB) Ltd

Spray, Turntables
Atlas Copco (GB) Ltd

Spraying Equipment, Metal Working
Atlas Copco (GB) Ltd
IMI Norgren Enots Ltd (Shipston)
Stewart-Warner Ltd

Spraying Installations (See also Guns): Automatic
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
IMI Norgren Enots Ltd (Lichfield)

Spraying Installations, Paint
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd

Starters, Air Motor – For Diesel Engines
Ingersoll-Rand Sales Co Ltd

Stud Setters
Fiam Utensili Pneumatici SpA

Switches, Pressure and Vacuum, Electric (See also Alarm Warnings)
CompAir Maxam Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
Martonair Ltd
Mecman Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader-Bellows Ltd
Systematic Design & Developments Ltd
Thomas Wright (Bradford) Ltd

Tampers, Backfill
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd

Tappers, Thread
The Aro Corporation (UK) Ltd
Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd

Thrusters
CompAir Maxam Ltd
Mecman Ltd

Timers, Delay
CompAir Maxam
Crouzet Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Systematic Design & Developments Ltd

Timers, Electrical (Dial, Sequence, Delay, Process, Automatic – Hand reset)
Crouzet Ltd
R.G.S.Electro-Pneumatics Ltd

Timers, Electro-Mechanical (Dial, Sequence, Delay, Process, Automation and Hand reset)
Crouzet Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd

Timers, Pneumatic (Dial, Sequence, Delay, Process, Automatic and Hand reset)
CompAir Maxam Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Systematic Design & Developments Ltd

Tool Steels, Chisel Steels
Atlas Copco (GB) Ltd

Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Padley & Venables Ltd

Tool Steels, Digger Steels
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Padley & Venables Ltd

Tool Steels, Dressing Tools
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Padley & Venables Ltd

Tool Steels, Feathers
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Padley & Venables Ltd

Tool Steels, Moil Steels
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Padley & Venables Ltd

Tool Steels, Pick Steels
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Padley & Venables Ltd

Tool Steels, Pile Drivers
Consolidated Pneumatic Tool Co Ltd
Padley & Venables Ltd

Tool Steels, Plug and Feathers
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Padley & Venables Ltd

Tool Steels, Punches
Atlas Copco (GB) Ltd
Ingersoll-Rand Sales Co Ltd
Padley & Venables Ltd

Tool Steels, Spades
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Padley & Venables Ltd

Tool Steels, Tamping Pads and Stems
Atlas Copco (GB) Ltd
Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Padley & Venables Ltd

Torque Actuators
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd

Traps, Water (See also Filters and Separators)
Consolidated Pneumatic Tool Co Ltd
Domnick Hunter Filters Ltd
Mecman Ltd
Pall Pneumatics Ltd
Thomas Wright (Bradford) Ltd

Tyre Inflators
Thomas Wright (Bradford) Ltd

Vacuum Cleaners, Pneumatic
IMI Norgren Enots Ltd (Shipston)

Vacuum Pumps
Charles Austen Pumps Ltd
CompAir Maxam Ltd
Dawson McDonald & Dawson Ltd
Howden Engineering Ltd
Ingersoll-Rand Sales Co Ltd
Joy Manufacturing Co (UK) Ltd
Lacy-Hulbert & Co Ltd
Northey Rotary Compressors Ltd
Thomas Wright (Bradford) Ltd

Vacuum Tweezers
Charles Austen Pumps Ltd

Valve Actuators
CompAir Maxam Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
R.G.S. Electro-Pneumatics Ltd
Schrader Bellows Ltd

Valves, Compressor, Suction and Discharge
Compressor Accessories Ltd
Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Mecman Ltd

Valves, Directional Control (for controlling pneumatic cylinders, etc)

The Aro Corporation (UK) Ltd
CompAir Maxam Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Drallim Controls Ltd
Festo Ltd
Fluid Automation Systems SA
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Valves Keyboard Selection

Systematic Design & Developments Ltd

Valves, Manual

CompAir Maxam Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Drallim Fluid Components Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
IMI Norgren Enots Ltd (Shipston)
Jouvenel et Cordier
Legris Ltd
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Valves, Mechanical

CompAir Maxam Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
Fluid Automotion Systems SA
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Thomas Wright (Bradford) Ltd

Valves, Needle (See also Regulators, Flow)

The Aro Corporation (UK) Ltd
CompAir Maxam Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Drallim Fluid Components Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
IMI Norgren Enots Ltd (Shipston)
Jouvenel et Cordier
Legris Ltd
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Thomas Wright (Bradford) Ltd

Valves, Non-Return or Check

NEI-APE Ltd Bellis & Morcom
CompAir Industrial Ltd
CompAir Maxam Ltd
Drallim Fluid Components Ltd
Compressor Accessories Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
IMI Norgren Enots Ltd (Shipston)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Valves, Pressure Reducing (See also Regulators)

The Aro Corporation (UK) Ltd
CompAir Maxam Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Jouvenel et Cordier
Legris Ltd
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd

Pnumatics Controls Ltd
R.G.S. Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Williams & James (Engineers) Ltd
Thomas Wright (Bradford) Ltd

Valves, Quick Exhaust

CompAir Maxam Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Valves, Relief Pressure

CompAir Maxam Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Northey Rotary Compressors Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Valves, Relief, Vacuum

Festo Ltd
Midland Pneumatic Ltd
Northey Rotary Compressors Ltd
Parker-Hannifin (UK) Ltd
Schrader Bellows Ltd

Valves, Remote

CompAir Maxam Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
Fluid Automation Systems SA
IMI Norgren Enots Ltd (Shipston)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd

Valves, Remote Control

CompAir Maxam Ltd
Davis Pneumatic Systems Ltd
Drallim Controls Ltd
IMI Norgren Enots Ltd (Shipston)
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics

Valves, Rotary

Crouzet Ltd
Drallim Controls Ltd
Festo Ltd
Martonair Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics

Valves, Safety

Crouzet Ltd
Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Martonair Ltd
Midland Pneumatic Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Thomas Wright (Bradford) Ltd

Valves, Sequence

CompAir Maxam Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd

Valves, Shuttle

CompAir Maxam Ltd
Crouzet Ltd

Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Valves, Solenoid Operated

The Aro Corproation (UK) Ltd
CompAir Maxam Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Valves, Solenoid Operated: Explosion Protected

R.G.S.Electro-Pneumatics Ltd

Valves, Solenoid Operated, Flameproof

Davis Pneumatic Systems Ltd
Festo Ltd
Fluid Automation System SA
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Martonair Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Sperry Vickers Automation & Pneumatics
Thomas Wright (Bradford) Ltd

Valves, Speed Control (See also Regulators, Flow)

CompAir Maxam Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Festo Ltd
IMI Norgren Enots Ltd (Lichfield)
Jouvenel et Cordier
Legris Ltd
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Thomas Wright (Bradford) Ltd

Valves, Stop and Shut-Off

CompAir Maxam Ltd
Crouzet Ltd
Davis Pneumatic Systems Ltd
Drallim Fluid Components Ltd
Festo Ltd
IMI Norgren Enots Ltd (Shipston)
Jouvenel et Cordier
Legris Ltd
Martonair Ltd
Mecman Ltd
Midland Pneumatic Ltd
Parker-Hannifin (UK) Ltd
Pnumatics Controls Ltd
R.G.S.Electro-Pneumatics Ltd
Schrader Bellows Ltd
Thomas Wright (Bradford) Ltd

Vibrators, Concrete Immersion

Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd

Vibrators, External

Consolidated Pneumatic Tool Co Ltd
Thomas Wright (Bradford) Ltd

Vibrators, Internal

Consolidated Pneumatic Tool Co Ltd
Ingersoll-Rand Sales Co Ltd
Thomas Wright (Bradford) Ltd

Winches

Ingersoll-Rand Sales Co Ltd
Joy Manufacturing Co (UK) Ltd

Wire Brush Machines

Ingersoll-Rand Sales Co Ltd
Stewart-Warner Ltd

Sub-section (c)

ALPHABETICAL LIST OF MANUFACTURERS WITH ADDRESSES, TELEPHONE AND TELEX NUMBERS, OF HEAD OFFICE, WORKS AND BRANCHES

THE ARO CORPORATION (UK) LTD, Alders Drive, Redditch, Worcs B98 ORH
Telephone: 0527 25722 Telex: 338169

ATLAS COPCO (GB) LTD, P.O.Box 79, Swallowdale Lane, Hemel Hempstead, Herts.
Telephone: 0442 61201
Branches:
Wardpark South, Cumbernauld, Glasgow, Scotland.
Telephone: 02367 33722
Mansfield Road, Aston, Nr.Sheffield, Yorks.
Telephone: 0742 872831

CHARLES AUSTEN PUMPS LIMITED, 100 Royston Road, Byfleet, Weybridge, Surrey KT14 7PB
Telephone: Byfleet 43224/5

BP OIL LIMITED, BP House, Victoria Street, London SW1E 5NJ
Telephone: 01 821 2000. Telex: 8811151

BRISTOL PNEUMATIC LIMITED, Lodge Causeway, Fishponds, Bristol BS16 3JS
Telephone: 0272 657511 Telex: 44641

COMPAIR INDUSTRIAL LIMITED, P.O.Box 7, High Wycombe, Bucks.
Telephone: 0494 21181 Telex: 837371
Branches:
P.O.Box 44, Ipswich. Suffolk.
Telephone: 0473 217901 Telex: 98254

COMPAIR MAXAM LIMITED, Pool, Redruth, Cornwall TR15 3PR
Telephone: 0209 712712 Telex: 45433
Branches:
Arundel Road, Uxbridge, Middx UB8 2RR
Telephone: 0895 52571 Telex: 935197
London Road, Thrupp, Nr. Stroud, Gloucester GL5 2BA
Telephone: 0453 882863 Telex: 43454
Lightning Way, off Alvechurch Road, West Heath, Birmingham B31 3PH
Telephone: 021 477 7577 Telex: 337716
44 Melford Court, Harswich Grange, Warrington WA1 4RZ
Telephone: 0925 811610 Telex: 627592
91–93 Leigh Street, Attercliffe, Sheffield S9 2PR
Telephone: 0742 446481 Telex: 54537
71 Colvilles Place, Kelvin Industrial Estate, East Kilbride
Telephone: 035 52 35663 Telex: 777780

COMPRESSOR ACCESSORIES LTD, 1649 Pershore Road, Stirchley, Birmingham B30 3DR
Telephone: 021 458 1124/5/6 Telex: 338835

CONSOLIDATED PNEUMATIC TOOL CO LTD, Fraserburgh AB4 5TE, Aberdeenshire
Telephone: Fraserburgh 3131

CROUZET LTD, 108 Hawley Lane, Farnborough, Hants GU14 8HY
Telephone: 0252 513211 Telex: 858612

DAVIS PNEUMATIC SYSTEMS LTD, Newnham Industrial Estate, Plympton, Plymouth
Telephone: 336421/4 Telex: 45657

DAWSON,McDONALD & DAWSON LIMITED, Compton Works, Ashbourne, Derbyshire DE6 1DB
Telephone: 0335 43184

DOMNICK HUNTER FILTERS LIMITED, Durham Road, Birtley, Co Durham, DH3 2SF
Telephone: 0632 405121 Telex: 537282 a/b DHFLTR
Branches:
Domnick Hunter Filter GmbH, D 4150 Krefeld, Postfach 3106, Germany
Telephone: 02151/36041 Telex: 8531166 dhf
John Gunderson, Rosavej 6, 3660 Stenlose
Telephone: 17 14 07

DRALLIM FLUID COMPONENTS LTD, Brett Drive, Bexhill-on-Sea, Sussex TN40 2JP
Telephone: 0424 216611 Telex: 952585

J. H. FENNER & CO LTD, Fluid Power Division, Ashton Road, Harold Hill, Romford, Essex
Telephone: 040 23 43851 Telex: 21381

FESTO LTD, Catherine Wheel Road, Brentford, Middlesex.
Telephone: 01 568 2051/2/3/4 Telex: 935705 Grams: Festo Brentford
Branches:
Chancery House, Lombard Street, West Bromwich
Telephone: 021 525 6288
Unit 9, Josephs Well, Wesgate, Leeds
Telephone: 0532 459402
Suite 20, Carron House, Tay Walk, Cumbernauld, Scotland
Telephone: 02367 36908/9
Unit BR9B, Connswater Industrial Estate, East Bread Street, Belfast BT5 5AP
Telephone: Belfast 731514
Festo International Office, Postfach 6040, D 7300, Esslingen 1, Berkheim, W.Germany
Telephone: 07 11 391/0

FIAM UTENSILI PNEUMATICI SpA, Viale Crispi, 123— 36100 Vicenza, Italy.
Telephone: 0444 562611

FLUID AUTOMATION SYSTEMS SA, 124 route de l'Etroz, CH 1290, Verson, Switzerland
Telephone: 55 26 91/55 14 93 Telex: 22194

HPC ENGINEERING LTD, Victoria Gardens Industrial Estate, Burgess Hill, W.Sussex RH15 9RQ
Telephone: 04446 41671 Telex: 87606

HOWDEN COMPRESSORS LTD, 133 Barfillan Drive, Glasgow G52 1BE
Telephone: 041 882 3346
Branches:
Weybridge Industrial Estate, Weybridge, Surrey KT15 2GR
Telephone: Weybridge 56631
Howden Anbergen, Haven, Goor, The Netherlands
Telephone: 05470 3155
21 Union Street West, Oldham, Lancs
Telephone: 061 652 2235
Rupprechtgasse 3, A2340 Modling, Vienna, Austria
Telephone: 02236 82445
Burton Corblin, BP No.1, Nogent-sur-Oise, 60101, Creil, France
Telephone: 455 12 21
Azeta S.A.R.L., Via Locatelic, 20124, Milan, Italy
Telephone: 63 20 01

IMI NORGREN ENOTS LTD,, Norgren Works, Shipston-on-Stour, Warwickshire CV36 4PX
Telephone: 0608 61676 Telex: 83208 (Norsec G)
Enots Works, P.O.B.22, Eastern Avenue, Lichfield, WS13 6SB
Telephone: 05432 54151 Telex: 338555 (Enots G)
Branches:
Unit 71, Jarvis Way, Gravelly Industrial Park, Tyburn Road, Birmimgham, W.Midlands B24 8TJ
Telephone: 021 328 2811 Telex: 339547
6 Station Lane Industrial Estate, Birtley, Tyne & Wear DH3 1DQ.
Telephone: 0632 405821 Telex: 537463
Forrest Street, Blackburn, Lancs BB1 3BB.
Telephone: 0254 53666/7
Raikes Lane Industrial Estate, Bolton, Gt.Manchester BL3 2NE
Telephone: 0204 33317 Telex: 635191
Unit 22, Endemere Road, Albion Industrial Estate, Coventry, W.Midlands CV6 5PY
Telephone: 0203 661239/661117
IMI Pneumatics Ltd, Fleming Way, Crawley, W.Sussex RH10 2SJ
Telephone: 0293 513441 Telex: 877611
Airdraulics Ltd, Nore Road, Dublin Industrial Estate, Finglas Road, Dublin 11
Telephone: 0001 300288
32 Jordanvale Avenue, Clydeside Industrial Estate, Glasgow, Strathclyde G14 OQY
Telephone: 041 954 5037 Telex: 777988
Tudor Works, Bradgate Street, Leicester, Leics LE4 OAS.
Telephone: 0533 29218
Unit 9, Central Trading Estate, Hawthorne Road, Bootle, Merseyside L20 6PF
Telephone: 051 922 0444/5
713—714 Tudor Estate, Abbey Road, Park Royal, London NW10 7XX
Telephone: 01 965 0711 Telex: 935409
Greengate Works, 2 New Bridge Street, Greengate, Salford, Manchester M3 7WP
Telephone: 061 881 2358
Ram Mill, Gordon Street, Chadderton, Oldham, Gt.Manchester OL9 9QN
Telephone: 061 620 2431 Telex: 669517
23 Orgreave Crescent, Dore House, Industrial Estate, Handsworth, Sheffield, S.Yorks S13 9NQ
Telephone: 0742 697626 Telex: 547403
P.O.Box 163, The Butts, Worcester, Hereford & Worcs WR1 3EN
Telephone: 0905 27686 Telex: 336008

INGERSOLL-RAND SALES COMPANY LIMITED, Bowater House, Knightsbridge, London SW1X 7LV
Telephone: 01 584 5070
Branches:
Chorley New Road, Horwich, Bolton
Telephone: 0204 66544
Professional Tools Group, Swan Lane, Hindley Green, Nr.Wigan,Lancs.
Telephone: 0942 57131
Inter-City House, 80 Oswald Street, Glasgow G1 4PL, Scotland.
Telephone: 041 204 0888
John F. Kennedy Drive, Bluebell, Dublin 12, Ireland
Telephone: Dublin 505934/505935

JOUVENEL ET CORDIER, B.P.312, 92506 Rueil Malmaison Cedex, France.

JOY MANUFACTURING COMPANY (UK) LIMITED, Peel Park Place, East Kilbride, Glasgow G74 5LN
Telephone: 03552 39121 Telex: 77368
Branches:
Burlington House, Chesterfield, Derbyshire S40 1SB
Telephone: 0246 37127 Telex: 54239

LACY-HULBERT & CO LTD, Richmond Road, Croydon, Surrey CR9 4EA
Telephone: 01 680 5226 Telex: 21598 LH CRO 9
Branches:
Lacy Hulbert & Co GmbH, Vencoer Strasse 5-7, 5000 Koun 1, West Germany.
Telephone: 010 49221 514184

LEGRIS LIMITED, Ashville Trading Estate, The Runnings, Cheltenham, Glos GL51 9PT
Telephone: 0242 43103/4/5 Telex: 43335

ING ENEA MATTEI SpA, Strada Padana Superiore 307 – 20090 Vimodrone, Milano, Italy
Telephone: 02 2500441

MARTONAIR LIMITED, 128 St. Margarets Road, Twickenham TW1 1RJ
Telephone: 01 892 4411 Telex: 935562
Branches
257 Summer Lane, Birmingham B19 2PX
Telephone: 021 359 6081/7
Unit 10, Wimbledon Ave, London Road Industrial Estate, Brandon, Suffolk IP27 ONZ
Telephone: 0842 811964
21 Ashton Vale Road, Ashton Gate, Bristol BS3 2HA.
Telephone: 0222 494551
Colchester Trading Estate, Colchester Avenue, Cradiff CF3 7AP
Telephone: 0222 494551
Martonair House, Carlisle Street, Gateshead, Tyne & Wear NE10 0LD
Telephone: 0632 692248 (3 lines)
84 Kilbirnie Street, Glasgow G5 8JD
Telephone: 041 429 7579 and 7570
9 Wallace Way, Hitchin SG4 OSD
Telephone: 0462 2243
Station Road, Southwater, Horsham, West Sussex RH13 7SW
Telephone. 0403 731171 and 731451
Unit 3, Penraevons Industrial Estate, Meanwood Road, Leeds LS7 2AW
Telephone: 0532 629338 (2 lines)
11 Cannock Street, Leicester LE4 7HR
Telephone: 0533 769993
28 Low Farm Place, Moulton Park Industrial Estate, Northampton NN3 1HY
Telephone: 0604 45793/4
James Dunlop (NI) Ltd, 26 Roughfort Road, Mallusk, Newtownabbey, Co.Antrim, BT36 8RE
Telephone: 023 13 44721
39 Victoria Road, Romford, Essex RM1 2LH
Telephone: 0708 42256
Unit 7, Sherdley Road Trading Estate, Sherdley Road, St.Helens, Merseyside WA9 5AA
Telephone: 0744 55296/7 (2 lines)
439/443 Penistone Road, Sheffield S6 2GR
Telephone: 0742 347666
Unit 43, South Hampshire Industrial Park, Totton, Southampton SO4 3SA
Telephone: 0703 860277/8
Bankfield Trading Estate, South Reddish, Stockport SK5 7PA.
Telephone: 061 477 5830 (4 lines)
Unit 11, Liddall Way, Haslemere Industrial Estate, Horton Road, West Drayton, Middx.
Telephone' 089 54 47841 (4 lines)
Hare Street, Bilston, Wolverhampton, West Midlands WV14 7JS
Telephone: 0902 402431/2/3

MECMAN LTD, Mecman House, Suttons Park Avenue, Reading, Berks RG6 1AZ
Telephone: 0734 664456 Telex: 848410

MIDLAND PNEUMATIC LIMITED, Patrick Gregory Road, Wednesfield, Wolverhampton WV11 3DZ
Telephone: Wolverhampton 732501 Telex: 337430

NEI-APE LTD BELLIS & MORCOM, Icknield Square, Birmingham B16 OQL
Telephone: 021 454 3531 Telex: 337507

NORTHEY ROTARY COMPRESSORS LTD, Alder Road, Parkstone, Poole, Dorset BH12 4BB
Telephone: 0202 741725 Telex: 41444

PADLEY & VENABLES LTD, Callywhite Lane, Dronfield, Sheffield S18 6XT
Telephone: 0246 413301 Telex: 54254

PALL PNEUMATICS LIMITED, Europa House, Havant Street, Portsmouth PO1 3PD
Telephone: 0705 753545 Telex: 86251

PARKER-HANNIFIN (UK) LTD, 6 Greycaine Road, Watford, Herts
Telephone: Watford 44377

PNUMATICS CONTROLS LTD, P.O.Box 18, Leighton Buzzard, Beds
Telephone: 0525 370735

PROCESS SCIENTIFIC INNOVATIONS LIMITED, Industrial Estate, Bowburn, Co.Durham, DH6 5AD
Telephone: 0385 770 550

R.G.S. ELECTRO-PNEUMATICS LTD, Church Street, Accrington, Lancs.
Telephone: 0254 32000 & 35015

TENNECO COLORTREND BV, Anderol Division, P.O.Box 3045, 6202 NA Maastricht, The Netherlands
Telephone: 010 31 43 628888
Branches:
Lawrence Industries, Anderol Division, Mitcham Industrial Estate, Streatham Road, Mitcham, Surrey CR4 8AP
Telephone: 01 648 2272

SCHRADER BELLOWS LTD, Walkmill Lane, Bridgtown, Cannock, Staffs.
Telephone: Cannock 2644 Telex: 336150 Schrad G

SERCK HEAT TRANSFER, Warwick Road, Birmingham B11 2QY
Telephone: 021 772 4353 Telex: 337341 339805

SPERRY VICKERS AUTOMATION & PNEUMATICS, Halesfield 6, Telford, Shropshire TF7 4LF
Telephone: 0952 58600 Telex: 35286
Sherbourne Road, Balsall Heath, Birmingham B12 9DJ
Telephone: 021 440 3501
6 Washington Road, Paisley, Glasgow
Telephone: 041 887 6281
68-70 Old Bedford Road, Luton LU2 7PA
Telephone: 0582 413023/4
102 Pendlebury Road, Swinton, Manchester M27 1AZ
Telephone: 061 793 5941/6336
109 Leigh Street, Sheffield S9 2PR
Telephone: 0742 449611/449702

STEWART-WARNER LIMITED, Tynemouth, Tyne & Wear, NE29 7UE
Telephone: 0632 590655

SYSTEMATIC DESIGN & DEVELOPMENTS LIMITED, Frizington Road, Frizington, Cumbria CA26 3QY
Telephone: 0946 810179

ULTRAFILTER LIMITED, 33 Sandy Way, Amington Industrial Estate, Tamworth, Staffs B77 4DS
Telephone: 0827 58234
Branches:
Ultrafilter GmbH Dusseldorf, Bussingstrasse 4, D5657 Haan 1, West Germany
Telephone: 0104921292012
Ultrafilter Ag, Thurgauerstrasse 72, Oerlikerhus, 8050 Zurich, Switzerland
Telephone: 01 320 0500

WATTS REGULATOR (UK) LTD, London Road, Thrupp, Stroud, Glos GL5 2BA
Telephone: 045 388 3401

WILLIAMS & JAMES (ENGINEERS) LTD, Chequers Bridge, Gloucester GL1 4LL
Telephone: 0452 28431 Telex: 43233

THOMAS WRIGHT (BRADFORD) LTD, Thorite House, Laisterdyke, Bradford BD4 8BZ
Telephone: 663471 Telex: 51674
Branches:
Thorite Pneumatics Ltd, Lowfield Road, Leeds LS12 6BS
Telephone: 444554/9 Telex: 557628
Thomas Wright (Huddersfield) Ltd, Long Lane, Moldgreen, Huddersfield
Telephone: 34245 Telex: 51674
Thorite Pneumatics Ltd, Churchill Road, Off Wheatley Road, Doncaster
Telephone: 25492 Telex: 557628

Index

A

Abrasive Tools 306
Absolute Pressure 248
ABS Pipes 153,154,174
Actuator Efficiencies 54
Actuators 54,195,221–223,259
  Bellows 221–222
  Rotary 222–223
Adiabatic Compression 11,13,73,74,79,113
Adiabatic Flow 20
Adjustable Stroke Cylinders 206
Adsorption Dryers 168,169
Adze 310
Aftercoolers 100,113,160
  Water-Cooled 95
Air Blasters 380–381
Air Blowers . . . . . *See* Blowers
Air Bottles 382
Airbreakers 380–381
Air Bubble Techniques 392–393
Air Compression 9
Air Consumption 55–56, 200,234
Air Cooling 51,98,114,116
Air Cushions 390–391
Air Cylinders 28,30,197–216,325
  Air Consumption 200
  Basic Types 197
  Cushioning 203–205
  Double-acting 198,199,260
  End Covers 211
  Energy Saving 215–216
  Finish 211
  Grades or Specification 209
  Graphical Symbols 411
  Heavy Duty 211
  Holding to Failure Position 48
  Light Duty 209–210
  Maintenance 280–281
  Materials and Construction 209
  Medium Duty 211
  Mounting Classification 213,214
  Mounting Methods 212–215
  Multi-position 265–267
  Operating Time 201
  Operation Speed 201–203
  Piston Rods 211
  Pistons 211
  Plastic 210
  Rod Bearings 212
  Rod Ends 214
  Rod: Piston Diameter Ratio 212
  Safety Requirements 43–44
  Shock Loads 203
  Single-acting 259
  Speed Control 202
  Spring-Return 281
  Tie-Rods 211,215
  Trouble-Shooting 280
  Typical Standard Size Geometry 210
  See also under specific types
Air Dosage 373–374
  System Requirements 374–375
Air Dryers . . . . . *See* Dryers and Drying
Air Films 390–391
Air Flow Measurement 250–255
  Alternative Tests 255
  British Standard Practice 253
  Calculation Method 254
  Equipment 254
  Methods of 250
  Nozzle Selection 251
  With Nozzle on Delivery Side 251,255
  With Nozzle on Intake 251
Air-hydraulic Circuits 268–270
Air-hydraulic Conversion 269
Air-hydraulic Cylinders 203,217–220
  Speed Control 218–219
  Typical Applications 217
Air-hydraulic Intensifier System 31
Air-hydraulic Pumps 269–270
Air Intake Filter 97
Air Intake Temperature 56
Air Leaks 54
Air-lift Pads 390
Air Lines . . . . . *See* Compressor Air Lines
Air Locks 178
Air Mains 53
Air Motors . . . 32,224–234,259,296,306,325,326
  Advantages over Electric Motors 328
  Air Consumption Figures at Maximum Power 234
  Axial Piston 228
  Characteristics 224
  Compared with Electric Motors 329
  Compressed Air Supply 232
  Continuous Operation 224
  Control of 232–233
  Diaphragm 230–231
  Exhaust 233
  Gear 231
  Impulse Turbine 229
  Linear Motors 234
  Low Speed High Torque (LSHT) 231
  Noise Reduction 233
  Performance Parameters 233
  Piston-type 32,227,296,326
  Power Characteristics 330
  Reliability 330
  Silencers 233
  Turbine 229
  Types of 225
  Vane-type 32,225–227,296,326,328
Airmovers 354–355
  Operating Principle 354
  Performance Characteristics 355
Air Pressure 31

Air Properties 5–14
Air Receivers 53,100,102–112,131,160,380
Capacity 109
Design Formula 107
Dished Ends 110
End Shape Thickness 110
Location 111
Shapes and Volumes 109
Storage Capacity 102–103
Typical Sizes. 103
Useful Capacity of 104
Variable Demand 103
Air Separation. 345
Air Springs. 331–336
as Isolators 336
Configurations. 333–334
Force Diagrams 334
Load Capacity. 334
Multi-convolution. 334
Operating Pressure 332
Performance. 334
Rolling Sleeve 334
Selection Guide 336
Spring Force. 331–332
Working Characteristics 332
Air Starters 328–330,380
Natural Gas 330
Operating Air Pressure 328
Operating on Medium other than Compressed Air 330
Portable Power Units 330
Standby 330
Tank Size 329
Air Suspension Units 390
Air Tools. . . . . *See* Pneumatic Tools
Air Traps. 139
Air/Water Vapour Mixture. 7
Alcohol Dosage 374
Alcohol Mist Dosage 374
Amagat's Law 16
Amaton's Law. 15
AND and OR Function. 39
Asphalt Breakers 297
Aspirators, Compressed-air. 343–344
Atmospheric Air 6
Atmospheric Pressure. 253
Aquasports. 382
Automation 33
Basic Degrees of. 33
Avagadro's Law 16

B

Bacteria Removal 172
Ball Float Trap 178
BCAS Publications 407
Bell Crank 31
Bellows. 247,333
Actuators 221–222
Bends, Equivalent Pipe Lengths 150
Bifurcated Coaxial Fan. 350
Blank-off Pressure. 358
Blowers. 152,347–353
Definition 70
Displacement Formula 348
Fan 347,348
Noise Problems 350
Performance 348
Performance Parameters 352
Radial-flow Fans 351
Specific Speed. 353,353
See also Roots' Blower
Blow Guns 308
Boiling Point of Water 385
Boolean Algebra. 42,271
Bourdon Gauge 244–246,255
Boyle's Law 15
Branch Lines. 143–145,154
Maximum Recommended Flow 143
Breakers 297
Air Consumption 297
Breathing Air 398–400
Filters 398–400
Impurity Levels 398
Industrial Systems. 398
Oil-mist Removal 398
Bubble Barrier. 392
Buffer Cylinder 205
Burrs 308

C

Carbon Dioxide Processing. 345
Carbon Dioxide Removal 170
Cascade Techniques. 42,271
CETOP Publications 405–406
CETOP Recommendations for Circuit Diagrams 430–432
Check Units 203,217
Check Valves 89,189,339
Chucks 315
Circuit Design 35,39
Logic 271
Polylog System 275–276
Circuit Diagrams
CETOP Recommendations for 430–432
Design of. 430–431
Circuits. 256
See also under specific circuit types
Circular Saws 315
Clamping Head 31
Clark's Law 15
Clutches,Adjustable. 311
Coalescing Filters 166–167

Coanda Effect 354
Cold-sterilization 172
Combination Filters 167
Composite Materials, Seals. 240–241
Compressed Air 158–173
  Breathing. *See* Breathing Air
  Cleanliness Degree Required. 165
  Contamination 158,178
  'Dry' 168
  High Pressure 376–381
  Moisture Content 161
  Safety Requirements 43
  Sterile 172
  Treatment Methods. 158
  Water Removal 168
Compressed-air Aspirators 343–344
Compressed-air Mains. 135–142
  Basic Geometry 138
  Branch Mains 139
  Dead-end Mains 135
  Design Flow Rate Effects 138
  Intervals Between Pipe Supports 140
  Leakage Rate 141–142
  Mains Fall 139
  Maximum Recommended Flow. 136,137
  Outdoor Air Lines 141
  Ring-main System. 135
  Sizing. 136
  Take-off Points 139
Compressed-air Pipe System 100
Compressed-air Volume 14
Compressibility 7
Compressibility Charts 8
Compressibility Factor 7,22
Compressible Fluids 17
Compressible Gas Flow. 17
Compression Couplings. 175
Compression Diagram 73
Compression Efficiency 12
Compression Fitting 175
Compression Ratio . . 9,12,14,54,79,113,249,362, 376,397
Compressor Acceptance Tests 119–123
  Adjustment of Results to Guaranteed Conditions 122
  General Information 119
  Guaranteed Performance. 119,120
  Measuring Points 122
  Permissible Deviations from Guarantee Conditions. 123
  Preliminary Tests 119
  Results Derived from Observations 120
  Test Layouts. 121
  Test Observations 120
  Tolerance on Capacity and Specific Energy Consumption 123
Compressor Air Line Fittings 174–178
Compressor Capacity 55
Compressor Characteristics Comparisons. .124,132
  Influencing Selection 129
  Tabulated Data 126
Compressor Controls 89–96
  Automatic Temperature Control 94
  Clearance Pocket Unloading 91
  Continuous Regulation. 93
  Discontinuous Regulation 132
  Monitoring Devices and Systems 92
  Protection 93
  Regulation 92
  Reverse Flow Regulation. 91
  Speed Regulation 93
  Stop--Start. 93
  Suction Valve Unloading. 91
  Summary of 96
  Temperature Regulations 93–95
  Three-step Unloading. 91
  Unloading 90
Compressor Cooling 113
  Aftercoolers 113
  Air Cooling 114,116
  Closed-circuit Water 94
  Cooling Water Requirements 115
  Diaphragm Cooling 114
  Injection Cooling 114
  Intercooling 114,116
  Interstage. 113
  System Design. 127
  Water Cooling 114,118
Compressor Diagram 76,77
Compressor Efficiency 74
Compressor Installation 97–101
  Centralized. 97
  Decentralized 97
  Delivery Side Installation. 99
  Discharge Side Arrangement. 101
  Foundations 99
  Location 97
  Recommendations 173
  Take-off Points 162
  Tuned Delivery Pipe 101
Compressor Intake 98
Compressor Life Assessment. 130
Compressor Lubrication 85–88
  Flood Lubrication 86
  Force-feed Lubrication. 87
  General Guide to 85
  Grease Lubrication 87
  Oil Injection 88
  Sight-feed Drip Oilers. 87
  Splash Lubrication 86
  Systems of 86
Compressor Performance. 73–81
  Guaranteed on Test. 119,120
Compressor Power Input Required 52–53
Compressor Regulation. 53

Compressors 59
Air-cooled 98
Altitude Effects 81
as Exhausters 386
as Measuring Device 142
Axial-flow 69,71
Capacity Ranges 125
Centrifugal 71,72,114
Comparative Efficiencies 127
Constant Speed 59
Cooling Vane 67
Cost Analysis 128
Definition 70
Demand Load 78
Depreciation 128
Diaphragm 68
Differential or Stepped Piston 60
Discharge Temperature 78,79
Displacement 59
Double-acting 60
Driver Cost 128
Dynamic 59,69,77,88,96
Full-load/No-load System 131
Graphical Symbols 410
Horizontal 60
Input Power Required in Relation to Altitude 80
Labyrinth Piston 62
Liquid Ring 68
Lobe Rotor 64,77
Lubricated 60,398
Lubrication 131
Maintenance 128,283–285
Maintenance Costs 130
Mobile 83
Mounting Requirements 128
Multi-stage 62,73,376,378,381
Noise Suppression 82–84,131
Non-Lubricated 60,62
Oil Changes 284
Oil-cooled Rotary Vane-type 68
Oil-flooded Screw 63,128,131
Oil-flooded Rotary 130
Piston Type 77,81
Portable 83–84
Positive Displacement 76,119
Power Consumption 131
Power Requirements 73,74
Pressure Regulation 89
PTFE Ring 400
Radial Flow 69
Reciprocating . . 59,60,62,75,86,96,97,127,128 130,131,283,284,376
Rotary 59,285
Rotary Screw 62,76,88,96
Rotary Vane 96
Screw 88,92
Single-acting 86,97
Single-stage 60,114
Specific Speed 77
Stationary 82,89
Summary of Characteristics 66
Temperature Effects 78,81
Time Costs 130
Trouble Shooting 284
Two-stage 60,127,128
Valve Design 132
Vane 65,76,87–88
Vertical 60
Vibration 131
Water Cooled 98
Working Time Between Overhauls or Replacement 130
Compressor Selection 124–132
Characteristics Infuencing 129
Cost Aspects 124
Generalized Performance Envelopes 124
Parameters Involved 124
Concrete Breakers 291,297,299,304
Condensers 360,362
Connection Diagram 264
Connectors 175
Constant Speed Control 53
Contractors' Tools 290,291,297–305
Rotary 304
Control Mechanisms, Graphical Symbols 423
Control Methods, Graphical Methods . . . 424–425
Control Systems 33
Design 35
Conversion Efficiency 54
Conveyors, Pneumatic . *See* Pneumatic Conveyors
Cooling System 50
Cooling Towers 95
Copper Pipes 153–156
Copper Tubing 144
Core Box Vibrators 317
Corrosion Effects 107
Cost Aspects in Compressor Selection 124
Cost-effectiveness 50,97
C-rings 239
Critical Area 24
Critical Flow Conditions 23
Critical Pressure 9,23,24
Critical Temperature 9,23
Cylinders . . . . . *See* Air Cylinders

D

Dalton's Law 15
Damping Ratio 336
d'Arcy Equation 17
Density 5,8
Dental Drills 308
Dew Point 168,169

Diaphragm Cylinders 207
Diaphragm Motors 230–231
Diaphragm Pressure Gauges 246–247
Die-grinders 307
Diggers 299
Displacement Demand 55
Divided Path 40,42
Double-acting Cylinder Control 46
Drains and Drain Valves 139,162–163,178
Drifters 303
Drills 309,310
Drive Losses 53
Dryers and Drying 168–172,373
Absorption-type 373
System Requirements 374–375
Dust Extractors 309

E

Efficiency 50,51,127
Ejectors 362
Elastomeric Bellows 333
Electrical Controls 35
Electric Motors 51,53
Electro-mechanical Sequencing,
Step-counter Method of 271
Electronic Pressure Gauges 246
Emergency Breakdowns 56
Emergency Power Systems 378
Emergency Stop Controls 44
Emergency Stop Sensors 44
Energy Costs 52
Energy Losses 53
Energy Recovery 98
Energy Saving 131
Pneumatic Cylinders 215–216
Energy Sources, Graphical Symbols . . . 419–420
Ethane Recovery 345,346
Ethylene Recovery 345
Exhausters, Compressors as 386
Exhaust Flow 367
Exhaust Port Flow Regulators 189
Expansion Chamber 367,268

F

Fail-safe Arrangements 49
Fail-safe Cylinder Circuits 46
Fan Blowers 347,348
Fans
Axial-flow 352,371
Bifurcated Coaxial 350
Noise Control 369–372
Noise Level and Frequency Content 371
Noise Parameters 371
Noise Spectra 371
Octave Band Sound Power Level 372
Propeller-type 352,353
Radial-flow 351
Sound Power Level Estimation 371
Sound Power Output 372
Festo Shift Register Method 36
Fettling 317
Fettling Grips 318
Filters and Filtration 158–160,163–164,281
Air Intake Filter 97
Breathing Air 398–400
Coalescing 166–167
Odour-removing 399
Pneumatic Conveyors 396
Toxic Gases 399–400
Fitting Resistance Coefficient 147
Flow Measurement, Air *See* Air Flow Measurement
Flow Rate 23,25
Flow Velocity 22,24
Fluidic Controls 39
Force Magnification 30
Foundry Equipment 317–319
Free Air Delivered (FAD) 53,55,74
Free Air Volume 14
Friction Factor 18,147

G

Gas Ballast 359–360
Gas Booster 378
Gas Bottles 378,380
Gas Constant 5,7
Gas Cylinders 325
Gas Drying 170
Gas Equation 5,7,9
Gas Laws 15,16
Gases
Physical Data 10
Pressure-temp-volume Relationship 5
Properties 5–14
Gauge Pressure 248
Gauges,Pressure . . . . . *See* Pressure Gauges
Gear Motors 231
Grafcet Diagram 38,42
Graphical Symbols 408
Compressors 410
Control Mechanisms 423
Control Valves 412–415
Cylinders 411
Directional Control Valves 413–415
Energy Sources 419–420
Joint Industry Conference (JIC) 428–429
Non-return Valves 416
Pressure Control Valves 416–417
Pressure Intensifiers 412
Pumps 410
Rapid Exhaust Valves 416

Reservoirs 421–422
Shuttle Valves 416
Standard Electronic 433
Supplementary Equipment 426
VDMA 427
Grinders 306–307,318
Grinding Wheels 306

H

Harmonic Motion 32
Heat-activated Dryers 171
Heat Exchange Rate 116–117
Heat Exchangers 113
Calculations 116
Heat Exchange Surface Area 118
Heatless Dryers 170
Heat Loss 113
Heat Recovery 99
Heat Regenerated Dryers 168,169
Heat Transfer Coefficient 117,118
High Energy Rate Forming 382
High Pressure Pneumatics 376–381
Pressure Ranges 376
Typical Applications 377
Hoists 325
Air Motor 326
Cylinder 325–326
Performance Figures 327
with Power Trolleys 327
Hydraulic Cylinders 203
Hydraulics 39
Hydro-check Cylinders 217

I

Impact Cylinders 208–209
Impact Wrenches 312–313
Impellers
Backward-curved Bladed 351
Fan-type 347,348
Forward-curved Bladed 351
Squirrel Cage 351
Impulse Turbine Motors . . 229–230,296,328,345
Incompressible Flow 22
Inspection, Scheduled Periods for 281
Instant Couplers 175
Intake Filters 158–160
Intensifiers 31,217,219–220
Graphical Symbols 412
Internal Combustion Engines 52,53
Interruptible Jet 340
Isentropic Compression 11,75
Isolating Valve 101
Isolators, Air Spring 336
Isothermal Compression 11–13,73–75
Isothermal Factor 12
Isothermal Flow 18–19

J

Jacket-type Mufflers 369
Jetflow Airmovers 354–355
Joint Industry Conference (JIC)
Graphical Symbols 428–429
Jolt Squeeze Machines 318

K

Karnaugh-Veitch Maps 42,271
K Factor 74,76,111
Killed Steels 108

L

Leakage Detection 280,281
Leakage Losses 54,141–142
Leakage Measurements 51
Let-down Systems 378–380
Lever Mechanism to Increase Stroke 31
Lifts 325
Limiting Length 20,21
Limiting Pressure 21
Limiting Temperature 20
Limiting Values 19
Limiting Velocity 20
Limit Switches 37
Linear Actuators 54,55
Linear Motions 31,259
Linear Motors 209,234
Liquefied Gas Recovery 345
Load Factors 55
Logic Circuit Design 39
Logic Circuit Devices 273
Logic Circuits 271–277
Pneumaid Systems 274
Pneumatic *vs* Electronic 277
Polylog System Design 275–276
Symbols 273
Logic Control Units 35
Logic Element Controls 39
Logic Functions 39,271,272
Low Speed High Torque (LSHT) Air Motors . 231
Low Temperature Pneumatics 373–375
Lubricants 85
Lubrication
Compressor . . . . *See* Compressor Lubrication
Maintenance 282
Methods 131
Lubricators, Air Line 297

M

Mach Number 17–20, 22
Maintenance 51, 278–285
  Air Cylinders 280–281
  Air Tools. 278–279
  Appliances 279
  Compressors 128, 283–285
  Lubrication 282
  Pipework. 282
  Precautions Before and After 279
  Scheduled 278,281
Mass Flow 23
Maximum Pressure 19
Mcleod Gauge 247
Mechanization. 28
  Basic Degree of 29
Micro-organisms. 172
Microprocessors 35
  in Logic Control Circuits. 277
Microswitches 35
Mining 380–381
Mining Tools. 290,291,303
Moisture Content of Saturated
  Compressed Air 161
Moisture Separator/Reducing Valve Unit . . . 140
Moisture Traps 139
Motor-powered Tools. 296
Movement Sequences. 37
Moving Part Logic (MPL) Elements. 271
Mufflers 369

N

Nibblers 314
Nitrole Rubber 242
Noise Control 97,131,367–372
  Air Motors 233
  Compressors 82–84
  Fan Blowers 350
  Fans 369–372
  Percussive Tools. 369
  Rock Drills. 370
  Tool Steels. 370
Noise Reduction 131
Nozzle Constants 251,253
Nozzle Dimensions 251,252
Nozzle Flow 24
Nozzle Selection for Air Flow Measurement. . 251
Nut Setters. 312
Nylon Elastic Spiral Hose 144
Nylon Tubing 143,144,154–157,175

O

Odours, Filtration. 399
Oil Contamination 178
Oil-free Compressors 166
Oil-mist Lubricator 164
Oil Removal 164–167
  Filters 166
Oil Vapour Removal 167
On–Off Devices. 272
Optimum Line Sizing. 53
Orifice Coefficient 152
Orifice Formulas 26
Orifice Loss 152
O-rings 237,238
Overburden Drilling Equipment. 302
Override Valve. 49

P

Paint Spraying. 140,320–322
  Cold 320
  High-pressure 320
  Hot 320,322
  Low-pressure 320
  Summary of Systems. 321
  Techniques. 320
  Transport (Feed) Systems 322
  Various Types of Paints and Finishes. . . . 321
Paper Filters. 159
Parallel Actions with Polylog 41
Percussive Tools. 289,306,318
  Noise Spectrum 369
Perforated Plate 251,252
Permeability of Elastomers 242
Plastic Pipes 154
Plastic Tubing 153
Pipe Bending. 156
Pipe Materials 153–157
Pipe Supports 157
Pipework, Maintenance. 282
Piston Motors 32,227,296,326
Pirani Gauge 247
Pneumaid System 274
Pneumatic Circuits 259–267
  Connection Systems 264
  'Over Designed' 261
  'Over Simplified' 261
  vs Electronic. 277
Pneumatic Conveyors. 394–397
  Air Losses 396
  Blow Tank 395
  Delivery Unit 395
  Feed 396
  Filters 396
  High Pressure Systems 395
  Low Pressure Systems 394
  Medium Pressure Systems 395
  Related Devices 396
Pneumatic Handling Systems 397
Pneumatics. 28,39
Pneumatic Sequence 39

Pneumatic Springs . . . . . *See* Air Springs
Pneumatic Tools 55,289–296
Air Consumption 292–296
Classification 289
Exhaust Systems 370
Maintenance 278–279
Motor-powered 296
See also Contractors' Tools; Mining Tools; Motor-powered Tools; Percussive Tools; Roadbreaking Tools; Rotary Tools; Workshop Tools; and under specific applications
Pneumatic Vice 315
PNEUROP 407
PNEUROP/CAGI Publications 407
PNEUROP Code 111

PNEUROP Publications 404–407
Poisson's Law 16
Polylog System 42, 275–276
Polythene Tubing 156
Polytropic Compression 12,73
Poppet Valves 48
Positioning Cylinders 205–206
Power Unit 28
PowRlock Clamping Head 31
Precooler 168
Pressure Determination at any Arbitrary Section 23
Pressure Drop . 53,105,106,137,138,140,143,144, 250,251,265,281
Basic Formula 146
Calculations 146–152
in Flexible Hose with Couplings Each End 151
Nomogram 147
Through Bends and Fittings 147
Through Flexible Pipes 150
Pressure Effects 51,376–381
Pressure Excesses 282
Pressure Failure 46
Pressure Gauges 244–249
Bourdon 244–246
Diaphragm 246–247
Electronic 246
Vacuum 247
Pressure Gradients 19
Pressure Intensifiers . . . . . *See* Intensifiers
Pressure Loss
in End Fittings 152
Through ABS Fittings 149
Through Steel Fittings 149
Pressure Ratings 156
Pressure Ratio 76,367
Pressure Regulator 89,322
Pressure Relief Valve 89
Pressure Sensors 93
Pressure Switches 44,46–48
Pressure Transducer 246
Pressure Units 248–249
Pressure Vessels 376,380
Pressure Vessel Steels 108
Programmable Logic Controller 277
Propane Recovery 345
Publications 404–407
Pulse Circuits 264–265
Pumps
Air 347
Air-hydraulic 269–270
Backing 356
Diffusion 360
Ejector 361
Fluid Jet 361–363
Graphical Symbols 410
Rotary Piston 359
Steam Ejector 361
Steam Jet 362
Sump 304–305
Vacuum . . . . . *See* Vacuum Pumps
Vane 359
Vapour Booster 361
Water-operated Air Ejector 363
PVC Hose 175
PVC Reinforced Hose 154
PVC Tubing 145,156

Q

Quarrying Tools 303
Quick-relase Couplings 175

R

Rack and Pinion 30
Racks to Increase Stroke Length 32
Radial Reaction Turbine 345,346
Rammers 299
Reaction-forced Compensation 187
Reaction Turbine 345,346
Reamers 312
Refrigeration 345
Refrigeration Dryers 168
Regeneration Control 171
Reinforced PVC Hose 145
Relative Humidity 6
Reservoirs, Graphical Symbols 421–422
Reservoir Size 47
Reset Characteristics 44
Reset Valve 47
Resistance Coefficients of Fittings and Bends 149
Resonance 97
Reverse Air Feed 204
Reynolds Number 18,147
Riveting Hammers 316
Riveting Machines 30

Road Breakers, Muffling 370
Roadbreaking Tools 291
Robots, Industrial 32
Rock Drills 291,300–303
  Noise Control 370
  'Wet' and 'Dry' 301–303
Roots' Blowers 64,77,347
Rotary Actuators 222–223
Rotary Files 308
Rotary Motion 30
Rotary Movements 32
Rotary Tools 295–296,303–304,306
Rotating Cylinders 223
Rubber Hose 153,155,156

S

Safety Checks 49
Safety Factor 107
Safety Requirements 43
Safety Valve 112
Sand Blasting Machines 318
Sanders 307–308
Sand Rammers 317,319
Saturation Pressure 6,8
Saws 310
Scaling Hammers 318–319
Screwdrivers 310–312
Screwed Connections 174
Seals 237–243
  Air Permeability 242
  Basic Requirements 237
  Composite 240
  Leakage 280
  Life and Performance 280
  Piston 241
  Replacement 281
  Rod 241,242
  Rubber 242
    Selection 242
  Special Sections 239
Self-propelled Drills 303
Semi-killed Steels 108
Semi-rotary Motion 30
Sensing Devices 33
Sensors 340–342
  Back-pressure 341
  Conical Jet 342
  Interruptible Jet 340
Separators 178,281
Sequential Control 33
Servo Cylinders 46–48
Shaft Sinking 300
Shank Grinders 304
Shape Factor 110,111
Shears 314
Shift Register Chain 37,39
Shift Register Module 35,39
Shock Wave Formation 367,368
Silencers 368
  Air Motors 233
  Single-chamber Exhaust 368
Sinkers 300
Skin Diving 382
Solderless Connections 174
Solenoid Pilot 193
Solenoids 191,193
Sorption 169
Sorption Dryer 169
Spaders 299
Spares, Stocks of 56
Specific Energy Requirement 51
Specific Heat 16
Speed Control Methods 202–203
Spray Painting . . . . . *See* Paint Spraying
Springs . . . . . *See* Air Springs
Stagnation Conditions 23
Stagnation Pressure 22
Stagnation State 21
Stagnation Temperature 21
Stainless Steel Pipes 153
Standard Atmosphere 80,249
Standard Atmospheric Pressure 249
Standards 403–404
Starters . . . . . *See* Air Starters
Sump Pumps 304–305
Supersonic Flow 22
Supplementary Equipment,
  Graphical Symbols 426
Swimming, Underwater 382
Switchgear, Air Blast 382
Steel Pipe 143,153,174
Step-counter Method of
  Electro-mechanical Sequencing 271
Sterilization Filters 172
Stone-working Tools 299
Stop and Start Control 53
Stop Devices 262–263
Stopers 303
Stop Movement 44
Stop Valve 49,112
Structural Steels 108

T

Tampers 299
Tandem Cylinders 206
Tappers 312
Technical Atmosphere 249
Temperature Determination at any Section . . . 23
Temperature Effects 373–375
Temperature Sensors 93
Thermodynamics, Laws of 16
Three-position Cylinders 206–207

Thrusters 207
Time Control 33
Time Delay 41,42,263–265
Time-delay Circuits 337–338
Time Factor 55
Timers 337–339
  Camshaft 338
  Piston/Air Bleed 337–338
Toggle Mechanism 30
Toggle Press 31
Tool Chucks 315
Tool Retainers 297
Tools . . . . . *See under* Specific Types
Tool Steels. Noise Control 370
Torque Control Ratchet Wrench 313
Torque Cylinder 222
Torque-limiting Devices 311
Torr (unit) 249
Toxic Gases, Filters 399–400
Transfer Machines 42
Tube Bending 156
Tube Expanders 314–315
Turbine Motors 229,296,308,328
Turbo-expanders 345–346
Turbo-machines 347
Two-hand Release 262

U

Ultra High Efficiency Filter 167
Units, Pressure 248–249
U-rings 238–239

V

Vacuum Cooling 385
Vacuum Gauges 247
Vacuum Measurement 247
Vacuum Measurements Units 249
Vacuum Packaging 385
Vacuum Pressure 248
Vacuum Pumps 356–363
  Blank-off Pressure 358
  Mechanical 358
  Performances 356
  Selection of 386
  Types and Characteristics 357
Vacuum Range 384
Vacuum Techniques
  Low Vacuum Applications 387–389
  Mechanical Processes 385–386
Vacuum Technology 384–389
Valve Actuation 192–194
  Air Actuation 194
  Methods of 192
  Standard Symbols 192
Valves 181–191
  Air-actuated 194
  Back-pressure 379
  Bleed 190
  Categories 181
  Check 89,189,339
  Confirugations 183
  Control 203
    Graphical Symbols 412–415
  Diaphragm 337
  Directional Control 181
    Graphical Symbols 413–415
  Equivalent Pipe Lengths 150
  Flapper 291
  Flow Control 188
  Glandless Spool 186
  Metering 203
  Non-return 189
    Graphical Symbols 416
  Plate 291
  Poppet 185
  Port Configurations 185
  Pressure Control 189–190
    Graphical Symbols 416–417
  Quick Exhaust 190
    Graphical Symbols 416
  Reducing 379,380
  Rotary Plate 186–187
  Safety 328
  Shuttle 188
    Graphical Symbols 416
  Size Designations 183
  Slide 186
  Snap-action 339
  Solenoid 191
  Solenoid Actuation 186,193
  Spool 185,186
  Standard Symbols 182
  Stop 49,112
  Sub-base or Manifold Mounting 183
  Sub-base Patterns 185
  Thimble or Tubular 291
  Three-position 262
  Timing 338–339
  Types 185
Vane Motors 32,225–227,296,310,326,328
Vapour Pressure 6
Variable Speed Control 53
VDMA Graphical Symbols 427
Vehicle Suspension 390–391
Velocity Determination at any
  Arbitrary Section 23
Venturi Silencer 98
Vibration isolation 335
Vibration Levels 131
Vitrified Bonded Wheels 306
Volumetric Efficiency 74,75,79

W

Water, Boiling Point of 385
Water Content of Air 6,7
Water Cooling 51,98,114,118
Water Separation 160
Water Vapour 6,8,160,168
Welded Construction 107
Welded Fittings 174
Welded-on Flanges 174
Weld Flux Scalers 319
Wheatstone Bridge Circuit 247
Winches 325
Wind Tunnels 381
Wood Borers 309,310
Woodworking Tools 310
Work Factor 55
Workshop Tools 290,306–316
Wrapped Ply Construction 156

Index to Advertisers

ALUP—Kompressoren GmbH 68D
The Aro Corporation (UK) Ltd XIII
Atlas Copco (GB) Ltd 56B
Charles Austen Pumps Ltd 436B
Belliss & Morcom 64A
BP Oil Ltd 86C
Bristol Pneumatic Ltd 436B
Bryan Donkin Co Ltd. 64B/C
CompAir Industrial (BroomWade) 124B
CompAir Ltd . . . . . Bookmark
CompAir Maxam Ltd 270A
Compressor Accessories Ltd 436A
Consolidated Pneumatic Tool Co Ltd 288B
Crouzet Ltd 270D
Davis Pneumatic Systems Ltd 216A
Dawson, McDonald & Dawson Ltd XV
Domnick Hunter Filters Ltd 162B
Drallim Controls Ltd 188A
Drallim Tube Couplings Ltd XVI
J.H.Fenner & Co Ltd XII
Festo Pneumatics Ltd. 182A/296A
Fiam SpA 288A
Fluid Automation Systems SA. 188B
Howden Compressors Ltd 60A
HPC Engineering Ltd 68C
Ingersoll-Rand Sales Co Ltd. 68A
IMI Norgren Enots Ltd. 170A
Joucomatic Controls Ltd VIII
Joy Manufacturing Co (UK) Ltd. 60D/124A
Lacy-Hulbert & Co Ltd. 384A
Legris Ltd X

Martonair Ltd. XI
Ing. Enea Mattei SpA XIV
Mecman Ltd. 216B
Midland Pneumatic Ltd XVI
Northey Rotary Compressors Ltd. 68B
Padley & Venables Ltd IX
Pall Pneumatics Ltd 158A
Parker-Hannifin. XIV
Pnumatic Controls Ltd 188B
Process Scientific Innovations 158B
R.G.S.Electro-Pneumatics Ltd XIII
Schrader Bellows. IV
Serck Heat Transfer 114A
Sperry Vickers, Automation & Pneumatics. VII
Stewart-Warner Ltd 296B
Systematic Design & Developments Ltd. 270C
Tenneco Colortrend BV 86A
Ultrafilter Ltd. 162A
Watts Fluid Power Ltd XII
Williams & James (Engineers) Ltd. 60B/C
Thomas Wright (Bradford) Ltd. 68D